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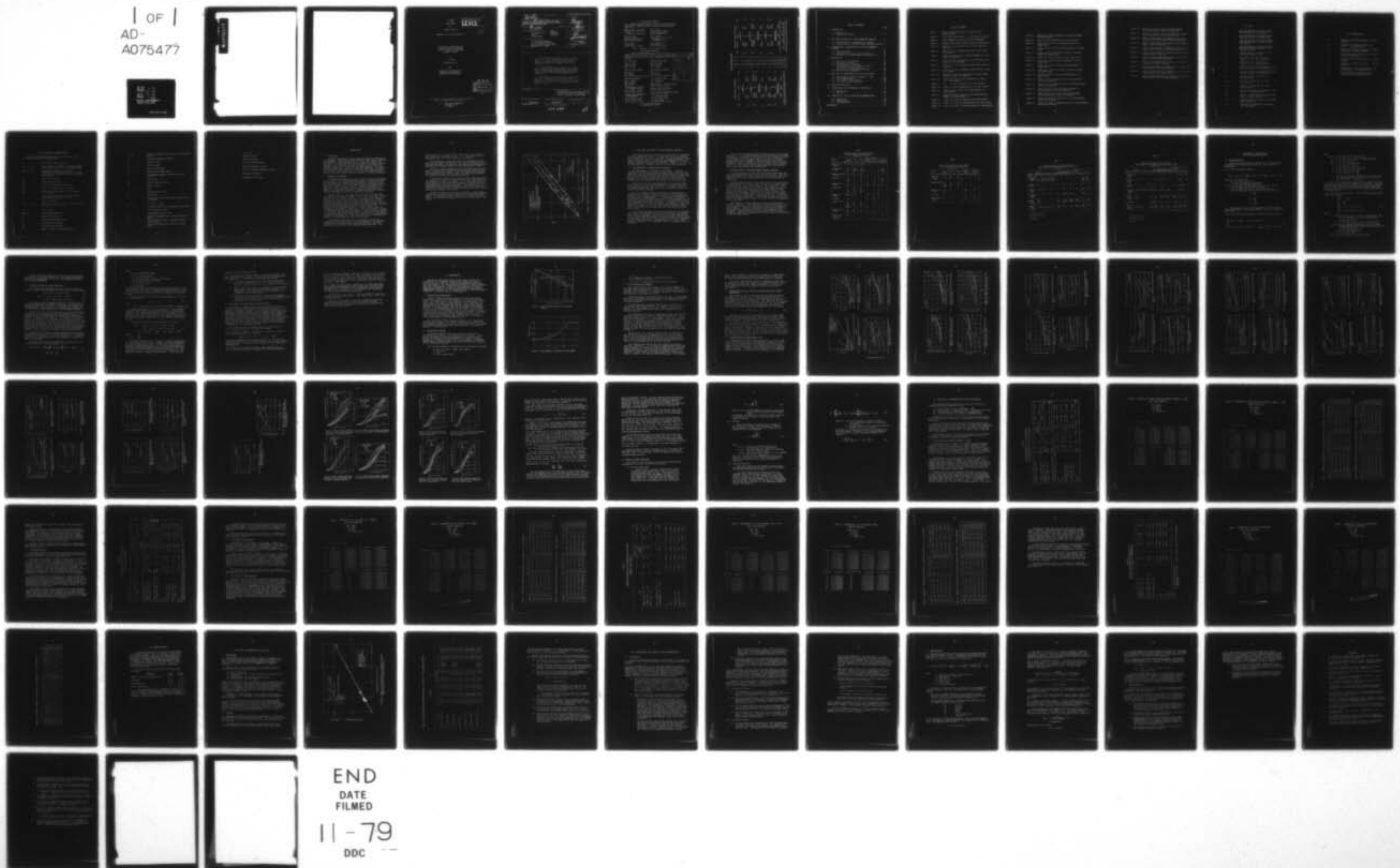

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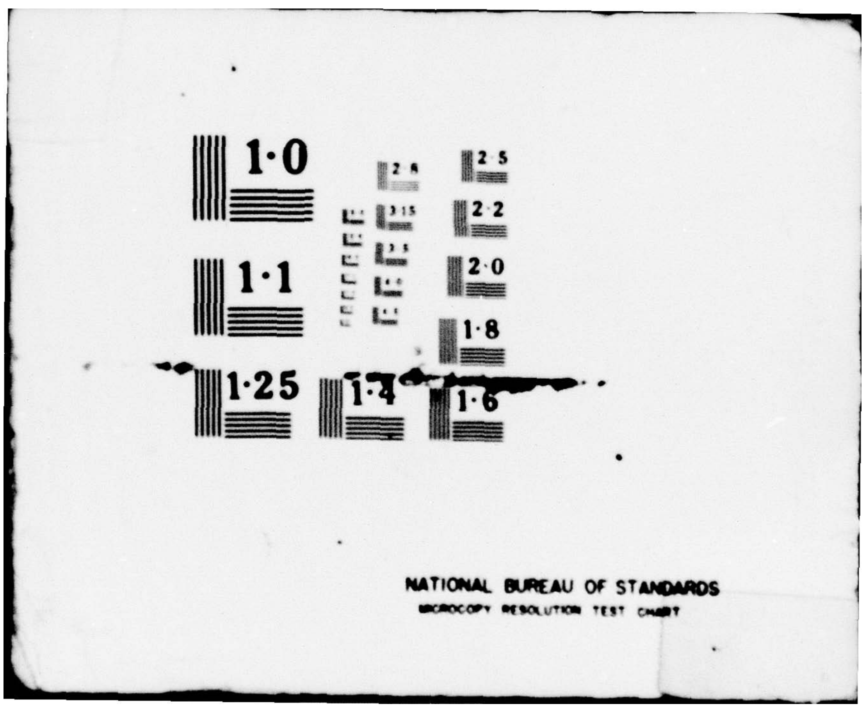
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FINAL REPORT

on

Project SR-1239

"Rational Limit of Hull Flexibility"

THE EFFECTS OF VARYING SHIP HULL PROPORTIONS AND HULL MATERIALS ON HULL FLEXIBILITY, BENDING AND VIBRATORY STRESSES

by

P. Y. Chang

Hydronautics, Inc.

under

Department of Transportation
United States Coast Guard
Contract No. DOT-CG-61906-A

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16. Abstract

The effect of varying ship proportions and hull materials on hull flexibility and on the concomitant bending and vibratory stresses for an ore carrier, a tanker, containership, and a general cargo ship is evaluated.

With the flexibility of the ship's hull represented by the natural frequency of the ship associated with the two-node shape, a potentially useful relation between the flexibility and bending moment has been established.

An analysis indicates that forward speed affects hydrodynamic damping and forces as well as hull flexibility, and there may exist an optimal flexibility for every ship, but there is not necessarily a limit to the flexibility.

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METRIC CONVERSION FACTORS

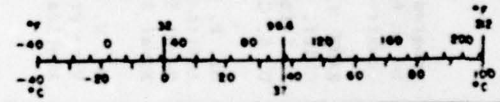
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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LIST OF SYMBOLS AND ABBREVIATIONS

The symbols and abbreviations defined in the text after the equations may not be repeated here.

A	Shear area of ship section.
A_{ai}, I_{ai}, Y_{ai}	Area, moment of inertia, and distance from the neutral axis of the i^{th} aluminum member.
A_{si}, I_{si}, Y_{si}	Area, moment of inertia of the i^{th} steel member, and the distance between its center of gravity and the neutral axis of the whole ship section.
B	Beam of ship section.
BM	Bending moment amplitude.
BM_L	Lateral bending moment amplitude.
BM_T	Torsional bending moment amplitude.
BM_V	Vertical bending moment amplitude (wave + still water).
C	Total damping coefficient of ship section associated with vertical motion.
C_B	Block coefficient
C_o, C_s, C_3, C_5	Damping coefficients/length as defined after the equations in the text.
D	Depth of ship.
Defltn	Ship maximum deflection.
E	Modulus of elasticity.
G	Shear modulus of elasticity.
g	Acceleration of gravity.
GM	Ship metacentric height.
I, I_2	Moment of inertia of ship section.

I_o, I_{my}	Mass rotary moment of inertia of ship section/length.
I^*	Equivalent moment of inertia.
L	Length of ship.
M	Bending moment
m_s	Mass of ship/length.
m_a	Added mass of water/length.
N	Hydrodynamic damping coefficient of ship section.
P	Axial force.
SM	Section modulus of ship.
T	Draft of ship.
t	Time.
U	Forward speed of ship.
V	Shear force.
w	Displacement, including deflection and rigid body motion.
w', $\dot{w}$	$w' = \frac{\partial w}{\partial x}$, $\dot{w} = \frac{\partial w}{\partial t}$
x	Coordinate along the longitudinal center line.
α	Heading angle of ship.
Δ	Displacement in long tons.
Δ_a	Added displacement due to the added mass of water.
θ	Fore and aft attitude, includes trim and elastic slope.
ζ	Wave surface elevation relative to still water.

μ	$= m_s + m_a$
ν	Poisson's ratio.
ρ	Density of water.
σ_b	Vertical bending stress.
ω	Natural frequency of ship.
ω_1	First or two-node frequency of ship.
ω_e	Encounter frequency.
ω_n	Frenquency of n^{th} mode.

1.0 INTRODUCTION

1.1 Objectives

Shipboard vibration has been a major problem for shipbuilders and operators. Vibratory stresses adversely affect ship structures and equipment, reduce fatigue life of a ship, and impair crew operations. At this time there are no generally accepted limiting standards or corresponding design procedures for assessing hull vibration, due in part to the lack of understanding of the relationship between ship proportions and hull vibration. Accordingly, the objective of this study is to determine the effects of ship proportions on hull flexibility and to establish suitable criteria for hull-vibration limits, such as a limit to the hull flexibility.

1.2 Summary of Findings

The methodology adopted for this study is based on two assumptions. First, it is generally believed that the existing methods for determining the seaway loads are adequate. Secondly, it is believed that ships with more flexibility are inferior to stiffer ships with respect to hull vibration. These two assumptions are generally accepted and are based on reliable information. For example, in 1970, Salvesen, Tuck and Faltinsen published their paper on sea loads(1), wherein the comparison between the analytical and experimental results are generally quite good.

Theoretically, for the same sea loads, more flexible ships are generally subjected to higher stress. For this reason, a more flexible ship is, indeed, inferior to a stiffer ship. However, study results reported herein differ considerably from these two assumptions. First, many shortcomings have been found in the existing methods of analysis and the corresponding errors indicate existing methodology may be inadequate for some problems. Secondly, results indicate the flexibility of the ship's hull is not necessarily an undesirable property. A more flexible ship can actually be safer than a stiffer ship. For these reasons, a limit to flexibility has not been established. From the results obtained in this study, the investigators tend to believe that there exists an optimal flexibility for every ship, but there is not necessarily a limit to the flexibility. This conclusion will be discussed in detail in the following sections of this report.

The primary study objective of determining the effects of variations of ship proportions on hull flexibility and vibratory responses for four ship types, have been achieved. The ship proportions are defined by two nondimensional parameters: The

length-beam ratio, L/B and $L^2/BI^{\frac{1}{4}}$. The effects of the depth, D, are included in the moment of inertia, I. The effects of the beam-draft ratio, B/T, were found to be negligible.

The flexibility of the ship's hull is represented in this report by the natural frequency of the ship associated with the two-node mode shape. An important and useful relation between the flexibility and bending moment has been established in Figure 1.

Because of the shortcomings of the existing methods of analysis, the qualitative values of these curves are more important than the quantitative values. Until these quantitative values are confirmed by more reliable input data and study methodology, the results presented are considered tentative.

In addition to studying the effects of the ship proportions, the study also achieved a broader goal of better understanding of the responses of ships in a seaway. It is clear that a more accurate method for ship-vibration analysis is required and can be developed within the state-of-the-art of the current theories of hydrodynamics and structural mechanics. For this reason a review of the existing theories and recommendations for new methodologies are included in this report.

During the course of the study, the effect of ship speed on damping was a subject of major concern and corresponding investigation. A tentative analysis indicates that forward speed has effects on hydrodynamic damping and forces as well as hull flexibility.

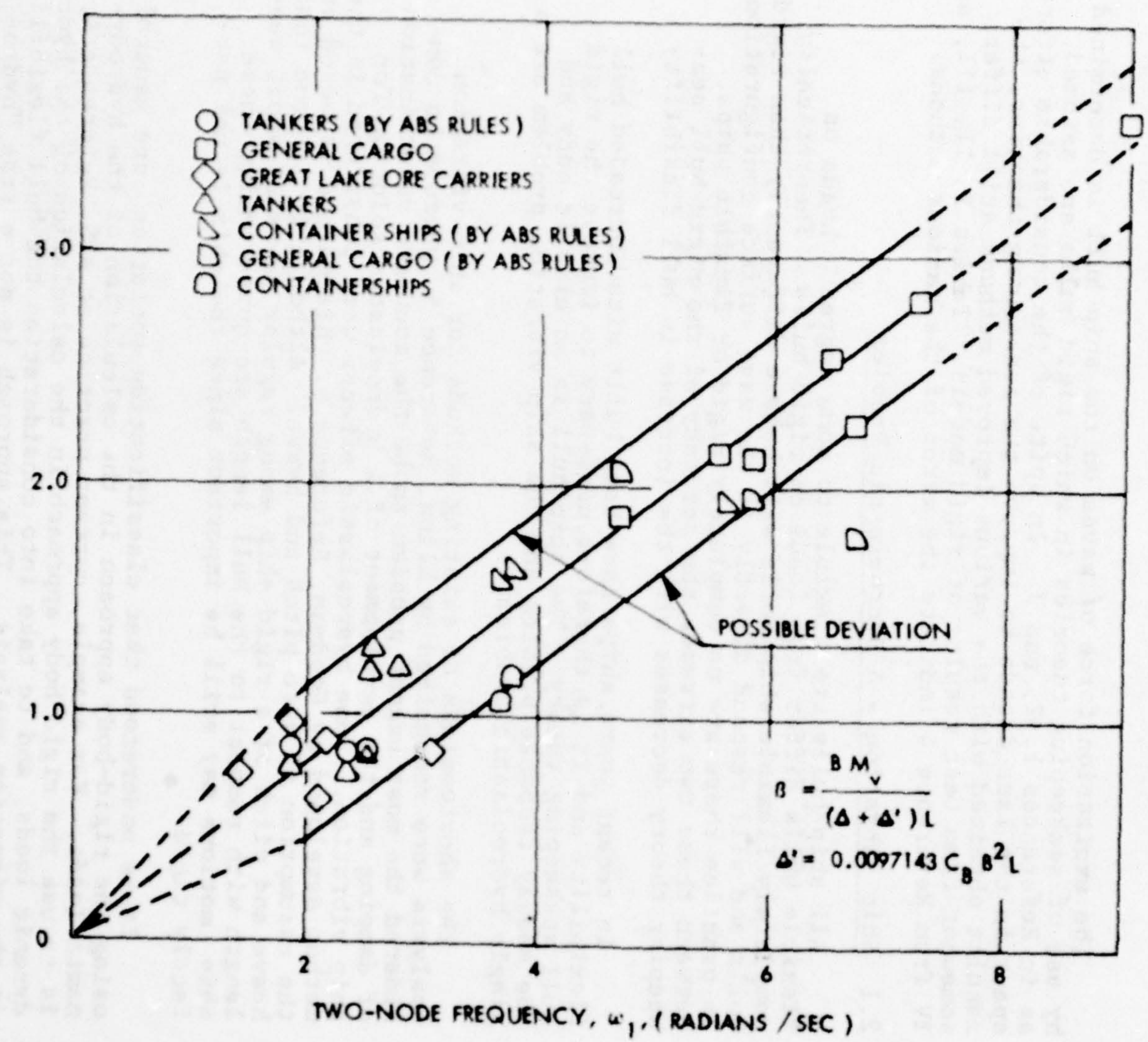

FIGURE 1 - EFFECT OF HULL FLEXIBILITY ON THE VERTICAL BENDING MOMENT

2.0 THE STATE-OF-THE-ART OF SHIP-VIBRATION ANALYSIS

The excitation force of waves on the ship hull is determined by use of seakeeping theories in which rigid hulls are assumed, as in References 1, 2, and 3. In spite of the considerable effort spent in the last decade to improve the seakeeping theories, the results obtained with the various improved methods still differ somewhat from test results of rigid models. Tables I, II, III, and IV from Reference 4 indicate the error of the various methods.

2.1 Ship Vibration - A Hydroelastic Problem

All ship hulls are flexible to some degree. Loads on flexible hulls differ from loads on rigid hulls. Theoretically, completely flexible ship hulls will behave differently than rigid hulls and will respond directly to the wave surface configuration. In practice there are no completely rigid or flexible ships. Between these two extremes, the accuracy of the rigid-hull seakeeping theory decreases with the increase in hull flexibility.

In recent years, ships have been built with increased hull flexibility and it is, therefore, necessary to improve the rigid hull seakeeping theory. The ship hull is an elastic body and the seaway response problem and the ship vibration problem are a single hydroelastic problem.

The shortcomings of existing methods for ship-vibration analysis were recognized by Kline, Reference 5, wherein he considered the most urgent problem to be the accurate determination of damping and the development of a hydroelastic solution for ship vibration. Some hydroelastic effects were considered in the method developed by Goodman, Reference 6. His method is based on the assumption of zero pitch and heave. Although it is true that heave and pitch of a rigid ship among regular waves of short wave length with respect to the hull length are quite small, these ship motions may still be important since the ship is not perfectly rigid.

It is understood that classification societies are generally using the rigid-body approach in the calculation of the hydrodynamic loads. For example, current practice at ABS, Reference 8, is to use the rigid-body approach in the calculation of the hydrodynamic loads, and to take into consideration the hull flexibility in the vibration analysis. This approach is not a true "hydroelastic" formulation of the problem, since the coupling effect is neglected.

The effects of forward speed have been recognized by Hoffman, Reference 7, to be quite important in his investigation with model experiments. The subject is discussed later in a separate section of this report. It is interesting to note here the sizeable discrepancies between Goodman's theoretical results and the experimental results. Hoffman was able to explain some of the discrepancies. From the equations of motion given in the following section, it can be shown that Goodman neglected some important terms, which may explain the discrepancies.

2.2 Problem Areas in Existing Seaway Response Analysis

In comparison with results from rigid-model experiments the rigid-ship seakeeping method is not entirely accurate. The errors shown in Tables 1 -- 4 are in addition to the errors due to the flexibility of the ship hull and the sum of the errors may be significant.

Despite great progress in the prediction of the seaway loads of rigid-ship hulls in recent years, two sources of error remain to be corrected. First, strip theories are, in general, valid only for the mid-body of the ship hull. The theory is not valid for the hull ends and errors tend to increase toward the ends. Since the effect of the forward speed is proportional to the changes of hydrodynamic coefficients, with great changes toward the ends, the accumulated errors can be significant. In recent years, efforts have been made to improve the accuracy of the added mass and damping coefficients. A promising approach is the use of finite-element methods wherein all types of hull cross-sections can be considered.

The effects of forward speed are another unsettled area. Salvesen, Reference 1, has indicated that the forward-speed terms in the equations of motion developed by various investigators differ greatly. From a brief review of the various versions of the forward-speed effects, Salvesen's version appears to be acceptable. However, additional studies and comparisons should be made to identify the importance of various terms in the analysis of forward-speed effects.

TABLE I

Approximate Measures of Correlation Between Theory and Experiment for Head Seas

Percent Error

Source	Froude Number	Pitch	Heave	Midship Vertical Moment	Midship Vertical Shear	Relative Bow Motion
Baitis, et al (1974)	.13 –.2	5-10	10-20	-	-	-
Cox and Gerzina (1975)	.22	5-10	5-15	-	-	5-10
	.30	10-15	5-15	-	-	5-30
	.37	20	10-30	-	-	5-30
Baitis and Wermter (1972)	.15	10	10	-	-	-
	.46	40	20	-	-	-
Flokstra (1974	.22	-	10	-	-	-
	.245	10	10	10	20	10-15
	.27	-	10	-	-	-
Wahab and Vink (1975)	.15	5	-	10	15	15
	.245	15	25	15	20	25
Journee (1976)	.15	10	20	-	-	-
	.20	10	25	-	-	-
	.25	10	25	-	-	-
	.30	10	20	-	-	-
Kaplan, et al (1974)	.25-.30	10-15	-	30	20	-
Kim (1975)	.25	-	-	10	30	-
Loukakis (1975)	.15	10	10	-	-	-
	.20	15	10	-	-	-
	.25	15	10	-	-	-
	.30	15	10	-	-	-
	.09-.14	10	10	-	-	-
Salvesen, et al (1970)	.2	5	5	-	-	-
	.45	20	10	-	-	-
	.15	-	-	10	10	-
Oosterveld and van Oossanen (1975)	.3 –.4	-	-	-	-	10

TABLE 2

Approximate Measures of Correlation Between
Theory and Experiment for Following Seas
Percent Error

Source	Froude Number	Pitch	Heave	Midship Vertical Moment	Midship Vertical Shear
Baitis and Wermter (1972)	0.15	10	15	-	-
	0.46	150	80	-	-
Journee (1976)	0.15	10	5	-	-
	0.20	20	10	-	-
	0.25	15	10	-	-
	0.30	15	15	-	-
Kaplan, et al (1974)	0.25 - 0.30	15	-	60	80
Kim (1975)	0.25	-	-	25	15
Wahab and Vink (1975)	0.15	5	-	25	100

TABLE 3

Approximate Measures of Correlation Between
Theory and Experiment for Bow Seas (Headings 120 to 150°)
Percent Error

Source	Froude Number	Pitch	Heave	Roll	← Midship Moments →			Midship Vertical Shear	C_B	GM/B
					Vertical	Lateral	Torsional			
Baitis and Wermter (1972)	0.15	10-15	5-10	10-50	-	-	-	-	.486	12%
	0.46	30-60	10-20	25-60	-	-	-	-	.486	12%
	0.15	10	10	50	-	-	-	-	.486	6%
Salvesen, et al	0.15	10	-	-	15	15	20	15	.80	5
Flokstra (1974)	0.245	20	30	15	15	25	40	30	.598	3.6
Fujii and Ikegami (1975)	0.195	15	25	-	20	30-50	30-50	-	.6994	4.1
Kaplan, et al (1974	0.25 - 0.30	-	-	-	40	20-40	20-90	40-90	.56	2.5
	0.25 - 0.30	-	-	-	40	20-40	20-90	40-90	.56	5.0
Wahab and Vink (1975)	0.15	10	-	-	25	20	30	30	.80	5.0
	0.245	10-30	20-30	20	30-50	25	20	50-100	.598	3.6

C_B = Block Coefficient

GM = Metacentric Height

B = Breadth

TABLE 4

Approximate Measures of Correlation Between
Theory and Experiment for Quartering Seas (Headings 30 to 60°)
Percent Error

Source	Froude Number	Pitch	Heave	Roll	← Midship Moments → Vertical	Lateral	Torsional	Midship Vertical Shear	C_B	GM/B
Baitis and Wermter (1972)	0.15	10	10	10	-	-	-	-	0.486	12%
Salvesen, et al (1970)	0.15	10	-	-	15	20	20	-	0.80	5
Flokstra (1974)	0.245	15	15	90	10	25	-	30	0.598	3.6
Fujii and Ikegami (1975)	0.195	5-20	15-20	20-35	20-25	20-80	30-40	-	0.6994	4.1
Kaplan et al (1974)	0.25 - 0.30	-	-	90	50	30-100	10-50	60-80	0.56	2.5
	0.25 - 0.30	-	-	30	50	20-70	40-90	60-80	0.56	5.0
Kim (1974)	0.25	-	-	50-100	20-40	30-40	30-90	40-100	0.56	2.5
Wahab and Vink (1975)	0.15	10	-	-	20	50	30	100	0.80	5.0
	0.245	10-15	-	30-40	20-40	30-50	50-60	50-100	0.593	3.6

C_B = Block Coefficient

GM = Metacentric Height

B = Breadth

3.0 A HYDROELASTIC FORMULATION OF THE SHIP-VIBRATION PROBLEM

3.1 Existing Methods

The existing ship-vibration methods can be explained most conveniently by the equations of motion used by various investigators:

3.1.1 The fourth-order equation:

$$EIw'''' + \mu\ddot{w} + kw - \left(\mu\frac{EI}{AG} + I_o\right)\ddot{w}'' + C\dot{w} + \frac{I}{AG}\ddddot{w} = F(x,t) \qquad (1)$$

where w is the deflection
kw is the restoring force
C is the total damping coefficient
μ is the ship mass plus added mass
$F(x,t)$ is the vertical excitation force per unit length
A is the shear area of the ship section
I is the moment of inertia of the ship section
I_o is the mass moment of inertia of the ship section

$$\dot{w} = \frac{\partial w}{\partial t}$$

$$w' = \frac{\partial w}{\partial x}$$

$$\vdots \qquad \vdots$$

$$w'''' = \frac{\partial^4 w}{\partial x^4}$$

This equation, with slight variations, has been used by many investigators, including Noonan[10], Kline[5], McGoldrick and others.

3.1.2 The second-order equations, obtained from Reference 8:

$$\begin{cases} [EI_2 x_5']' - I_{my}\ddot{x}_5 - C_5\dot{x}_5 + K_3A_3G(x_3' - x_5) = 0 \\ \mu\ddot{x}_3 + C_3\dot{x}_3 + k_3x_3 + [EI_2x_5']'' - [I_{my}\ddot{x}_5 + C_5\dot{x}_5]' = F_3(x,t) \end{cases} \qquad (2)$$

where

$F_3(x,t)$ is the vertical excitation force

I_{my} is the mass rotary moment of inertia/length

k_3x_3 is the restoring force

K_3A_3 is the vertical shear area

I_2 is the moment of inertia

μ is the ship mass plus added mass

x_3 is the vertical deflection

x_5 is the rotation, $x_5 = \frac{\partial x_3}{\partial x}$

C_5, C_3 are the total damping coefficients associated with the longitudinal rotation and vertical motion of the ship section, respectively.

Note that in the above two equations the load on the ship hull is not a function of the deflection of the hull. The hydrodynamic forces are partially included in the terms associated with the added mass and the hydrodynamic damping coefficients. Most of the hydrodynamic forces due to the forward-speed effects have been ignored.

3.1.3 The first-order equations - The following equations were used in this study:

$$\begin{cases} w' = \theta + \frac{V}{GA} \\ \theta' = \frac{M}{EI} \\ M' = V + P\theta + I_O\ddot{\theta} + C_O\dot{\theta} \\ V' = m_s\ddot{w} + C_s\dot{w} + F(w,\zeta,x,t) \end{cases} \tag{3}$$

where

w, θ, M, V are the deflection, slope, bending moment, and shear responses of the hull, respectively

P is the axial force

I_O is the mass rotary moment of inertia/length

C_O and C_s are the damping coefficients per unit length associated with the rotation and vertical motions of the ship section

m_s is the ship mass/length

A is the shear area

$F(w,\zeta,x,t)$ is the vertical excitation force

Note in the above equation that the load on the ship hull is a function of the hull deflection. The differences between equation (3) and equations (1) and (2) are explained further in the following paragraphs.

3.2 Comparison Among the Existing Methods

For simple beams, Equations (1), (2), and (3) can be re-written as similar fourth-order equations in the following manner:

$$EIw'''' + C\dot{w} + \mu\ddot{w} + kw = F(x,t) \tag{4}$$

$$EIx_3'''' + C\dot{x}_3 + \mu\ddot{x}_3 + k_3x_3 = F_3(x,t) \tag{5}$$

$$EIw'''' + C_s\dot{w} + m_s\ddot{w} = F(w,\zeta,x,t) \tag{6}$$

In the above equations shear deflection and rotary inertia have been neglected for comparative purposes. Note that m_s is the mass of the ship only, while μ is the mass of the ship and the added mass. Similarly C_s is the internal ship damping, and C is the internal ship damping plus the hydrodynamic damping.

Equations (4) and (5) have been generally accepted by naval architects. These equations account for the effect of the surrounding water on the added mass and hydrodynamic damping in addition to the structural damping. This concept is not entirely correct. In Equation (6), the terms to the left of the equal sign do not include any consideration of surrounding water. This implies that the ship is moving as an elastic body and is excited by the surrounding water, and that the excitation force is a function of the waves and the deflection of the ship. Physically this concept is more realistic. Mathematically, it should lead to a more reliable solution of the ship vibration and ship motion problems. This is explained further in the following section.

3.2.1 Wave-Excitation Forces - Wave-excitation forces are still an unsettled subject among seakeeping investigators and various formulations are currently in use (see Reference 1). A full discussion of the relative merits of these versions is beyond the scope of this study.

From basic fluid mechanics theory the excitation force of the surrounding water can be expressed as:

$$F(w,\zeta,x,t) = -\frac{D}{Dt}\left(m_a\frac{D}{Dt}\ (w-\zeta)\right) - N\frac{D}{Dt}(w-\zeta) - \rho gB(w-\zeta) \tag{7}$$

$$\frac{D}{Dt} = \frac{\partial}{\partial t} - U\frac{\partial}{\partial x}$$

where

m_a is the added mass/length

U is the forward speed

N is the hydrodynamic damping coefficient

w is the deflection of the ship

ζ is the water surface

B is the beam

This expression simply states that the excitation consists of the inertia force (first term), the damping force (second term), and the restoring force. All of these force components are functions of the relative position between the water surface and the ship section.

For rigid-ship hulls the vertical displacement of a section can be expressed as

$$w = \eta_3 - x\eta_5 \tag{8}$$

where

η_3 and η_5 are heaving and pitching displacements, respectively.

Substituting Equation (7) into Equation (3), and combining these four first-order equations, we can obtain a fourth-order equation. This fourth-order equation is too complicated for comparison with the existing methods. For convenience, the shear deflection, the rotary inertia, the rotary damping, axial force, etc. are neglected and Equations (4), (5), (6) are used for comparison.

Substituting Equation (7) into Equation (6), we have

$$\begin{aligned} EIw'''' &+ (C_s + N - Um'_a)\dot{w} - 2Um_a\dot{w}' + m_aU^2w'' \\ &- U(N - Um_a')w' + \rho gBw + (m_s + m_a)\ddot{w} \\ &= m_a\ddot{\zeta} + (N - Um_a')\dot{\zeta} - 2Um_a\dot{\zeta}' - U(N - Um_a)\zeta' + \rho gB\zeta \end{aligned} \tag{9}$$

where

$$m'_a = \frac{\partial m_a}{\partial x}$$

Note that Equation (7) is just one of the many expressions for the wave-excitation forces. However, different expressions for wave-exciting forces result in different and less complete terms for forward-speed effects. In fact, the excitation forces from modern seakeeping theory based on incident and diffraction wave potentials are different from those in Equation (7). The significant factor is the absence in previous theories of some important terms which appear in Equation (9). Those terms are

discussed in the following section.

3.2.2 The Effects of Forward Speed - If the shear deflection and other properties are included, the above expression becomes much more complicated. The following conclusions can be drawn:

(1) The forward speed affects (1) the hydrodynamic damping and (2) the stiffness of the ship. The second effect has been generally ignored by ship vibration investigators.

(2) Some of the terms relating to damping due to the forward-speed effects have been ignored by many vibration investigators. The significance of this omission is considered in the following section.

(3) The terms ignored by the ship-vibration investigators have proven to be important by investigators concerned with flow-induced vibration of pipes and rods as shown in References 12, 13, 14, and 15.

3.3 The Effect of Forward Speed on Ship Motions

The effect of forward speed on ship motions has been of particular concern during the study and the subject has been considered in some depth to support the methodology adopted. Independent structural analyses of ocean thermal energy (OTEC) cold-water pipes, reported in Reference 12, provides some insight into the effects of water flow. Since the cold-water pipe problem also uses a set of equations of motions similar to Equation (3), the effects of the internal water flow are equivalent to the effects of the forward speed of the ships. In comparing the cold-water pipe solution with the methods used by various ship-vibration investigators, it is evident that some important terms have been ignored in the ship-vibration problem.

From Equation (9), the forward speed has three types of effects on the responses of the ship in a seaway:

3.3.1 Effects on damping - The effects on damping are shown in the following terms with the speed U:

$$\text{Damping force} = (C_s + N - Um_a{}')\dot{w} - 2Um_a\dot{w}'$$

The methods recommended by Goodman (Reference 6), and used by Hoffman (Reference 7) and Kline (References 4 and 5), have ignored the second term. This term is also neglected in the ABS method, (Reference 8).

3.3.2 Effects on the Hull Stiffness - The terms $m_a U^2 w''$ and $U(N - Um_a{}')w'$ have the effect of changing the natural frequencies

and the vibration responses. The first term $m_a U^2 w''$ can actually cause the resonance vibration of a pipe conveying fluid or solid rods in parallel flow. These two terms are entirely ignored in the usual ship-vibration analysis. The second term $U(N - m'_a)w'$ or its equivalent does exist in the seakeeping theory by Salvesen, Reference 1, and others.

3.3.3 Effects on Wave Loads - Physically, all terms associated with forward speed generate certain forces upon the ship's hull. Mathematically, the terms or their equivalents on the right of the equal sign of Equation (9), are defined as the wave loads, for comparison with the existing methods.

The terms $m_a \ddot{\zeta} + (N - Um_a')\dot{\zeta} + \rho g B \zeta$ are exactly the same as Goodman's solution (Reference 6). The terms, $2Um_a \dot{\zeta}'$, $U(N - Um_a)\zeta'$, have been ignored.

Again, it is necessary to note that a different version of the excitation will result in a set of different effects. However, all versions of existing methodology do indicate that many terms have been ignored.

4.0 METHODOLOGY

Because of the limited scope of this study, project calculations of the vibration response were carried out using existing methods. The wave loads and hydrodynamic coefficients were calculated by the program MIT5D developed by the Massachusetts Institute of Technology. Using the data obtained from this program, the vibration of the ship was calculated by the program BEAMRESPONSE, Reference 19, with modifications for handling damped vibrations, as shown in detail in Reference 21.

4.1 Selection of Sea Spectra

Figure 2 shows the assumed variation of wave peak energy frequency with significant wave height for ocean and Great Lakes waves. The lower curve is for representative ocean waves and the upper curve is for waves in Lake Superior. The ocean waves are represented by the Bretschneider spectrum having a peak energy frequency 10 percent greater than the well known Pierson-Moskowitz spectrum. The Bretschneider spectrum is probably more representative than the Pierson-Moskowitz for all ocean locations and is of more interest for the present study since the higher frequencies of wave energy will produce larger springing stresses. The Great Lakes waves are represented by the Jonswap spectrum which is based on analysis of available wave spectral data, with emphasis on the Lake Superior data from Reference 16.

For the baseline ships and the variations, the responses for different wave heights and different headings were calculated by the seakeeping program. The conditions associated with the maximum wave-induced bending moment were then adopted for the vibration analysis. See section 4.5.

4.2 Equations of Motion

As indicated earlier, the usual analytical methods for estimating ship vibration are not entirely satisfactory. Errors may be introduced in the usual assumptions of rigid hulls for seaway load estimates and flexible hulls for vibration analysis. Because of the limited scope of this study, however, the forward speed effects discussed earlier have only been partially accounted for as indicated in Figure 3.

The constant parameters in Equation (3) are defined as follows:

$E = 30 \times 10^6$ psi $= 1.9286 \times 10^6$ tons/ft^2
$G = E/2\,(1+\nu)$
$\nu = 0.3$
$P = 0$ (no axial force)

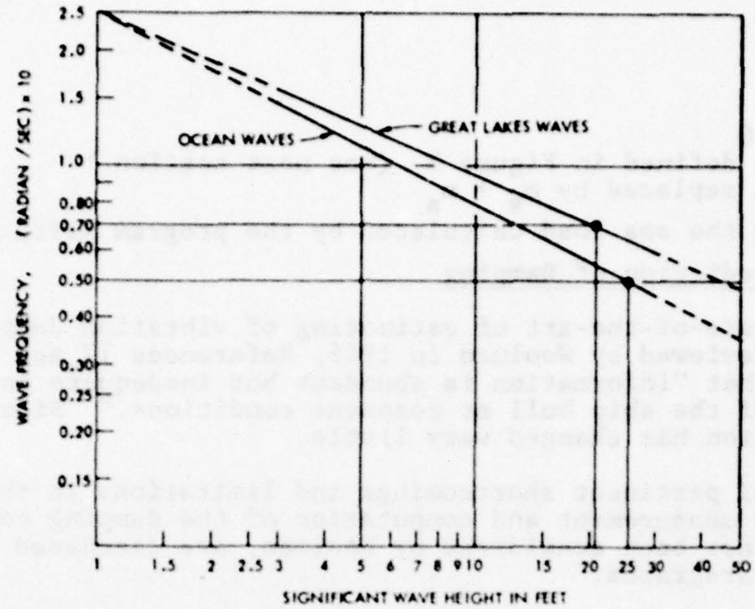

Figure 2 - Significant Wave Height VS. Peak Energy Frequency.

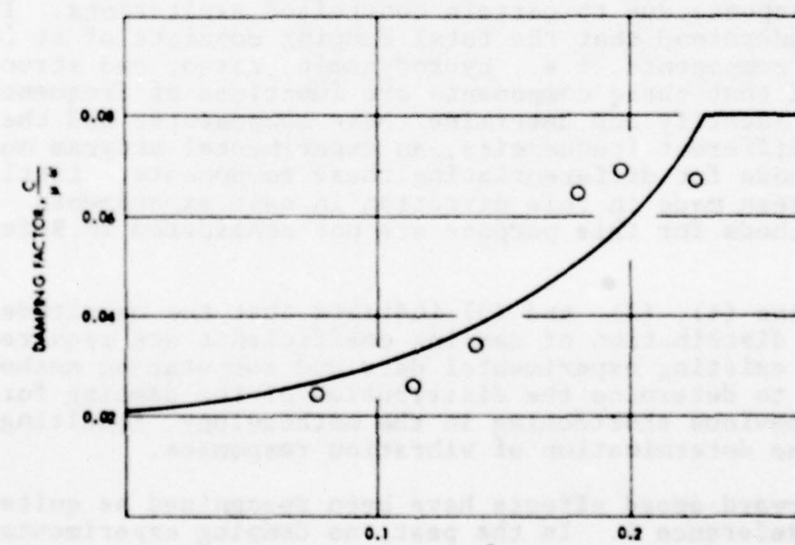

Figure 3 - Total Damping as a Function of Froude Number

$C_O = 0$
C is defined in Figure 3 (See next section.)
m_s is replaced by $m_s + m_a$
F is the sea load calculated by the program MIT5D

4.3 The Prediction of Damping

The state-of-the-art of estimating of vibration damping of ships was reviewed by Woolman in 1965, References 17 and 18. He concluded that "information is abundant but inadequate in predicting responses of the ship hull at resonant conditions." Since then, this situation has changed very little.

Several pertinent shortcomings and limitations in the existing methods for measurement and computation of the damping coefficients, which have not been considered by Woolman, are discussed in the following paragraphs.

Many measuring and computing methods treat the ship as a single damped mass-spring system. The results, even if accurate, provide the total damping of the ship. While such data are abundant and readily available, they are not adequate for ship vibration analysis.

The measurements obtained in damping experiments provide only the total response due to certain controlled excitations. It is generally understood that the total damping consists of at least three basic components, i.e., hydrodynamic, cargo, and structural damping, and that these components are functions of frequency. In order to identify and determine these components, and the effects of different frequencies, an experimental program must include methods for differentiating these components. Little effort has been made in this direction in past experiments. For example, methods for this purpose are not considered in References 17 and 18.

Equations (1), (2), and (3) indicate that the magnitude as well as the distribution of damping coefficients are required. None of the existing experimental data and computation methods can be used to determine the distribution of the damping force. This is an obvious shortcoming in the methodology, resulting in errors in the determination of vibration responses.

The forward-speed effects have been recognized as quite important, Reference 1. In the past, no damping experiments have been conducted to determine the forward-speed effects on damping. Hoffman, Reference 7, has calculated the difference between the experimental results and the results by Goodman's method, Reference 6, and he indicated the importance of the forward-speed effects. However, he attributed these differences to the damping

alone. Since Goodman's solution also ignores the forward-speed effects on the excitation force and the stiffness of the hull, the actual forward-speed effects on damping are still unknown.

The current indeterminate status of damping is considered in Figure 4. Various investigators use entirely different values of the damping coefficient. Note that almost, if not all, of these experimental data were measured with the ships being stationary.

4.4 Determination of the Effects of Ship Proportions on Hull Flexibility

The flexibility of any structure can be defined as the deformation of the structure at a given location produced by a unit generalized force, such as a deflection due to a unit force, and rotation due to unit moment, etc. This definition is not convenient for ships and its meaning is too vague for the designers. A better definition is the two-node frequency. It can be shown that ships with small values of vibration frequency respond to unit force with relatively great deformation. For this reason

$$\text{Flexibility} \sim \frac{1}{\omega_1} \tag{10}$$

The deflection due to a standard wave of unit wave height has also been used as a measure of flexibility. However, the result is not satisfactory because other factors such as heading angles and weight distribution have not been standardized.

The effects of varying hull proportions have been examined using non-dimensional hull geometry ratios such as L/B, B/T, $L^2/BI^{1/4}$, L/D, B/D, etc. The effects of these on bending moment, deflection, stress, and natural frequency of the hull have been plotted in Figures 5 through 37. From the study results, the effects of B/T were found to be quite small. Since the depth of the ship, D, does not affect the hydrodynamic force or the hydrodynamic coefficients, the effect of variation of D can be included in the structural moment of inertia of the ship, I.

4.5 Determination of Maximum Wave Loads

4.5.1 Maximum Wave Loads for High Energy Waves - In the study, the ship motions responses in a seaway were computed for a complete range of heading angles and different significant wave heights to determine the relation between the vertical bending moment and the wave heights. Some of the results are shown in Figures 38 to 45. The maximum wave height for the oceangoing ships is limited to 25 feet at a wave frequency of 0.5 rad/sec

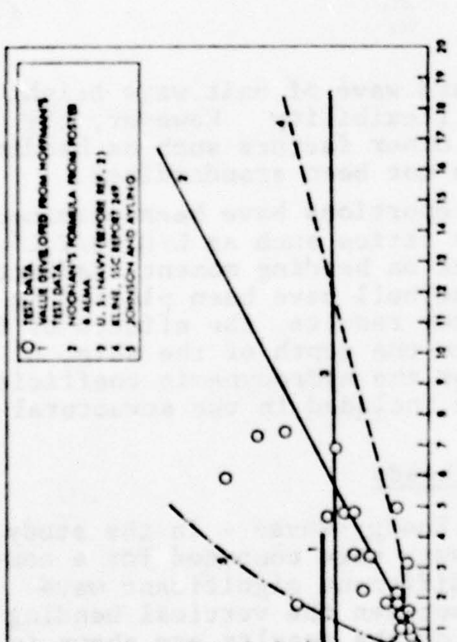

Fig. 4 - Damping Coefficients Used by Various Investigators

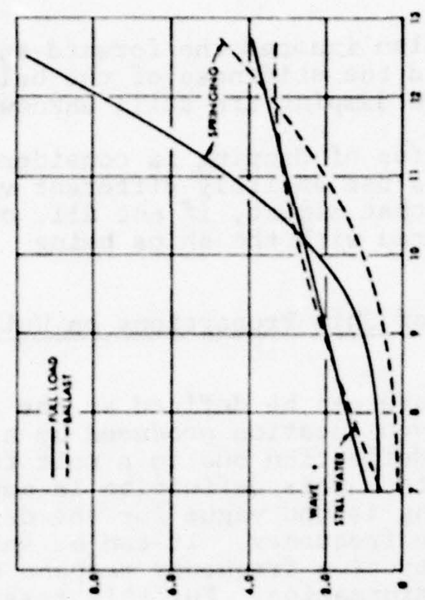

Fig. 5 - Effect of L/B on Deflection for Great Lakes Ore Carriers

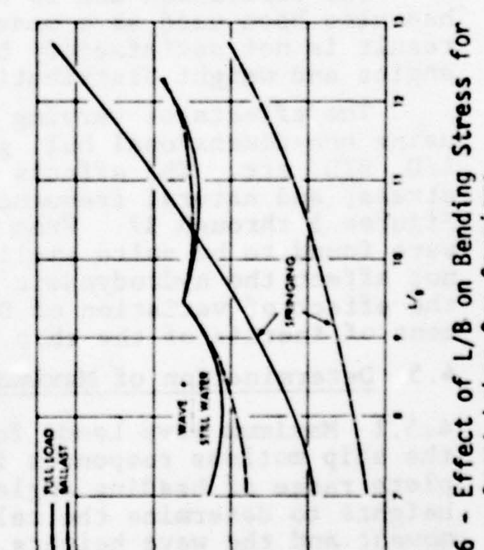

Fig. 6 - Effect of L/B on Bending Stress for Great Lakes Ore Carriers

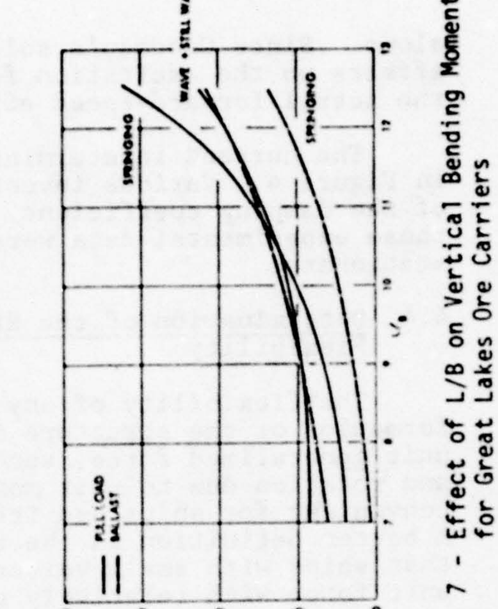

Fig. 7 - Effect of L/B on Vertical Bending Moment for Great Lakes Ore Carriers

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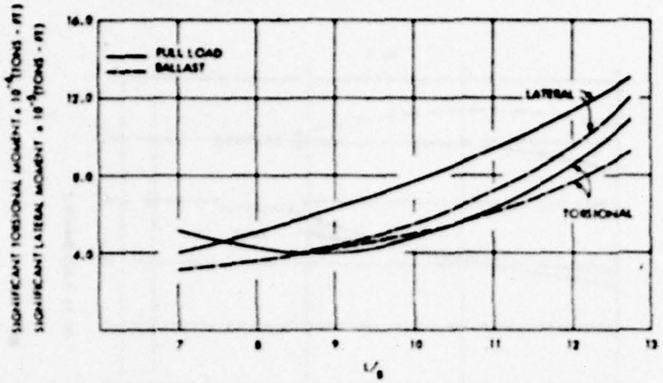

Fig. 8 - Effect of L/B on Lateral and Torsional Moment for Great Lakes Ore Carriers

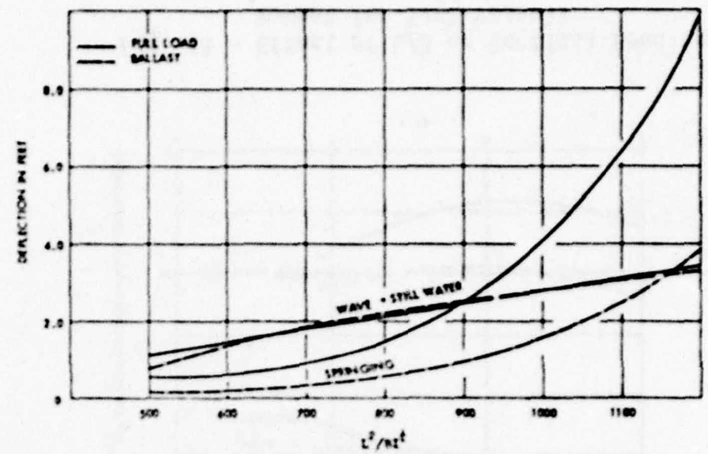

Fig. 9 - Effect of $L^2/BI^{1/4}$ on Deflection for Great Lakes Ore Carriers

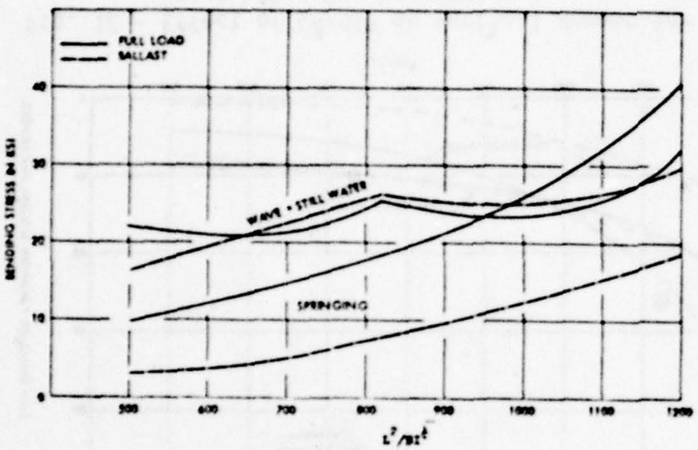

Fig. 10 - Effect of $L^2/BI^{1/4}$ on Bending Stress for Great Lakes Ore Carriers

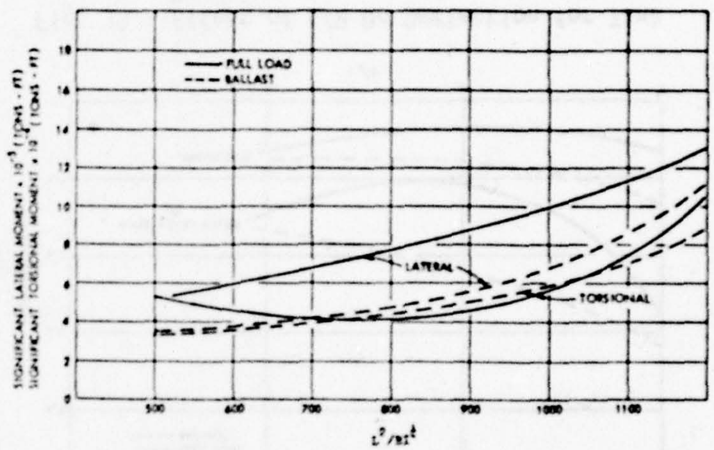

Fig. 11 - Effect of $L^2/BI^{1/4}$ on Lateral and Torsional Moment for Great Lakes Ore Carriers

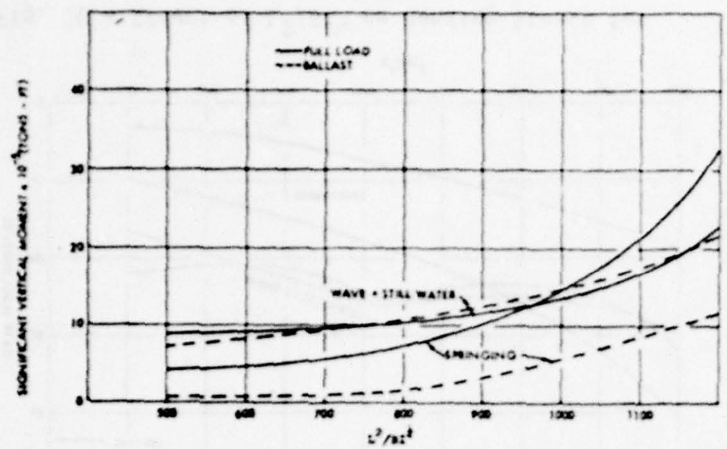

Fig. 12 - Effect of $L^2/BI^{1/4}$ on Vertical Moment for Great Lakes Ore Carriers

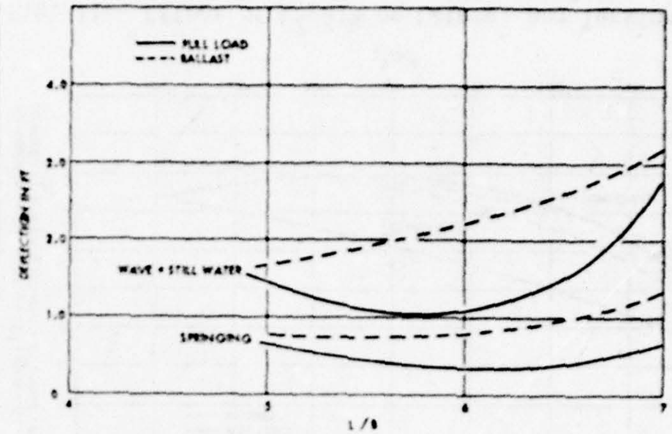

Fig. 13 - Effect of L/B On Deflection for Tank Vessels

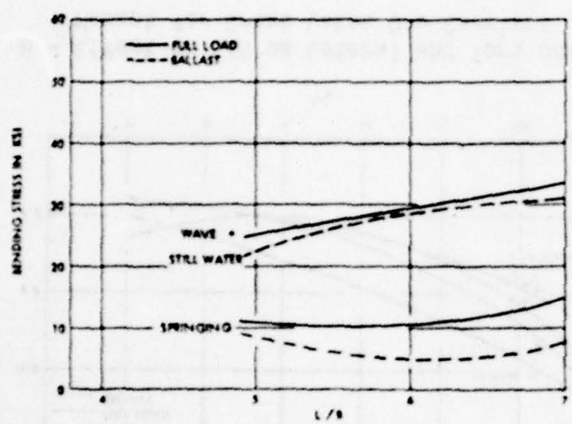

Fig. 14 - Effect of L/B on Bending Stress for Tank Vessels

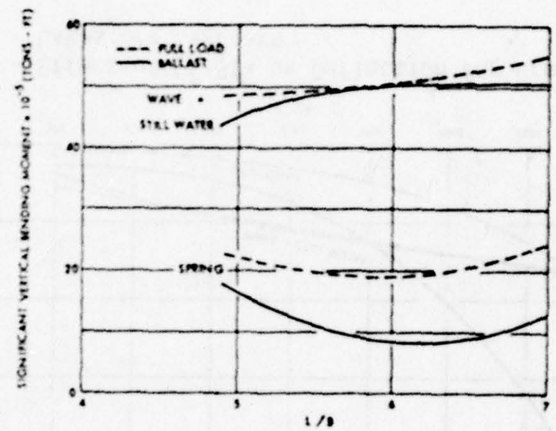

Fig. 15 - Effect of L/B on Vertical Bending Moment for Tank Vessels

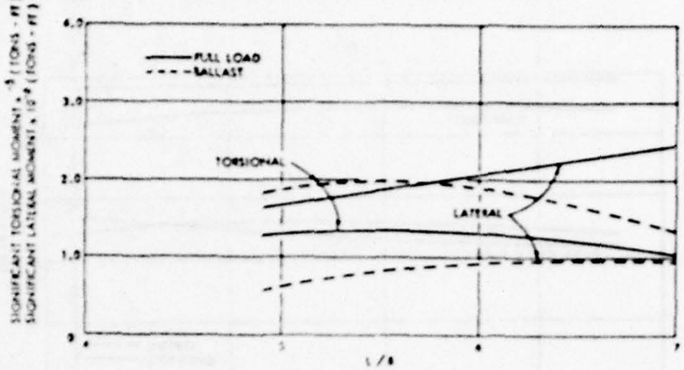

Fig. 16 - Effect of L/B on Torsional and Lateral Bending Moment for Tank Vessels

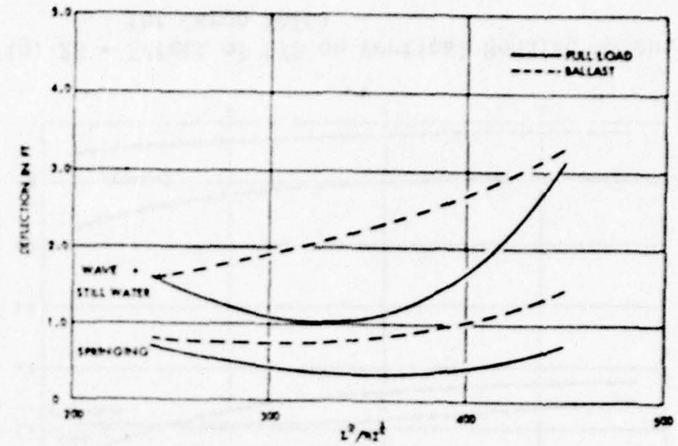

Fig. 17 - Effect of $L^2/BI^{\frac{1}{4}}$ on Deflection for Tank Vessels

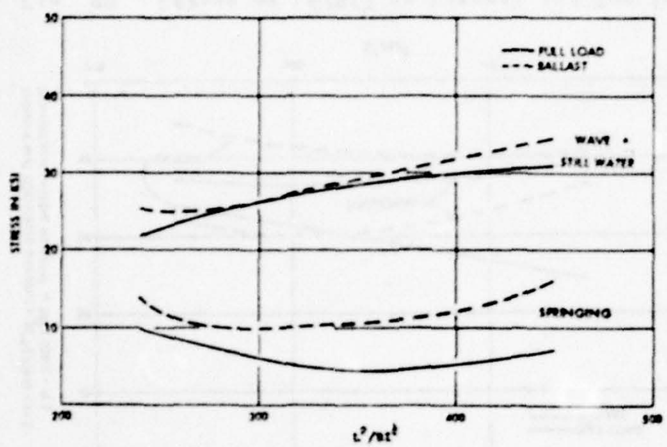

Fig. 18 - Effect of $L^2/BI^{\frac{1}{4}}$ on Bending Stress for Tank Vessels

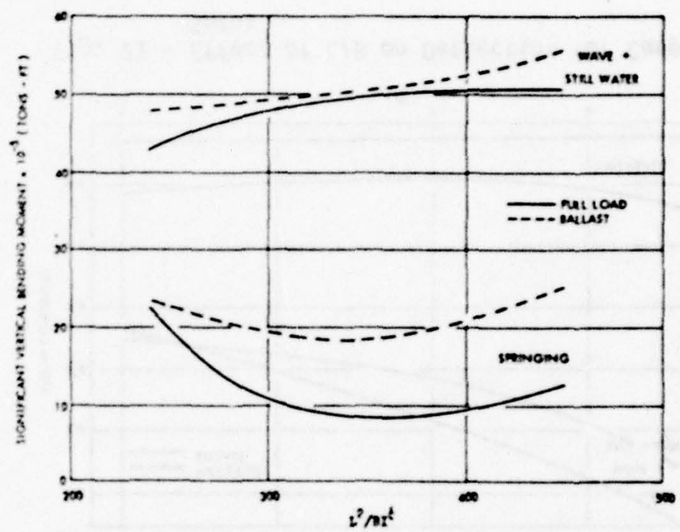

Fig. 19 - Effect of $L^2/BI^{\frac{1}{4}}$ on Vertical Moment for Tank Vessels

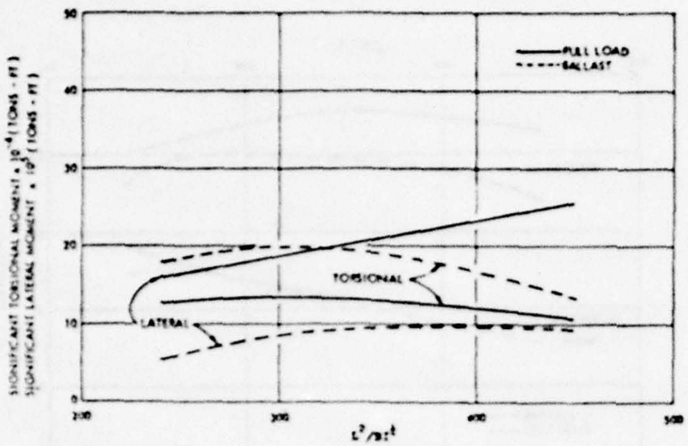

Fig. 20 - Effect of $L^2/BI^{\frac{1}{4}}$ on Lateral and Torsional Moment for Tank Vessels

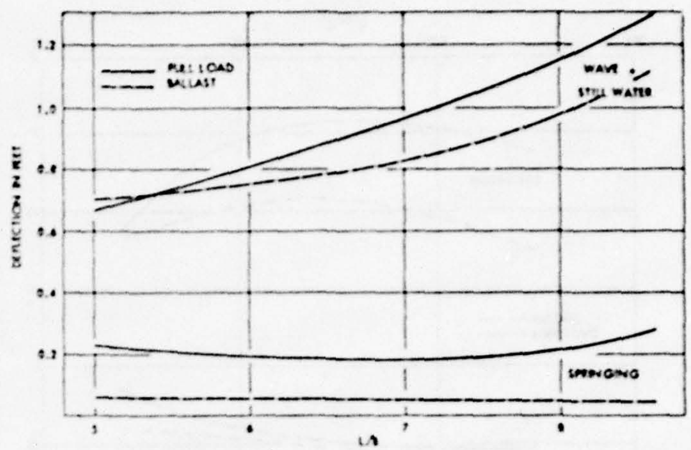

Fig. 21 - Effect of L/B on Deflection for Cargo Ships

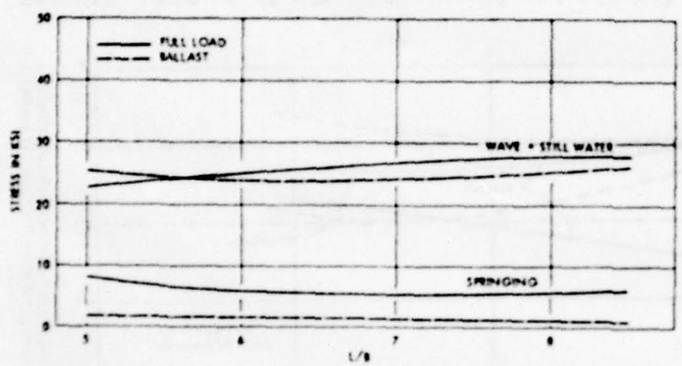

Fig. 22 - Effects of L/B on Vertical Bending Stress for Cargo Ships

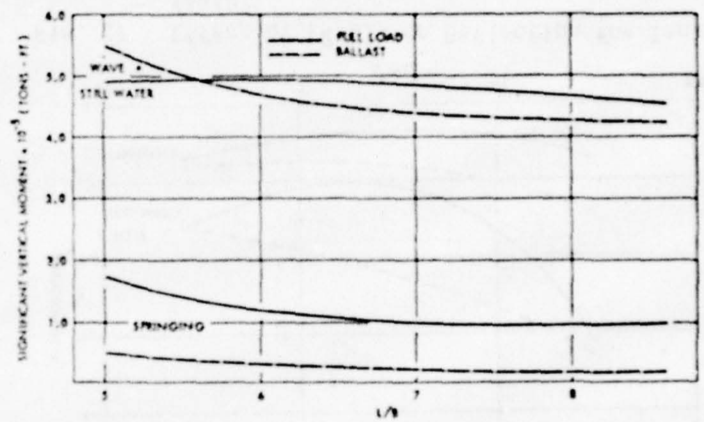

Fig. 23 - Effect of L/B on Vertical Bending Moment for Cargo Ships

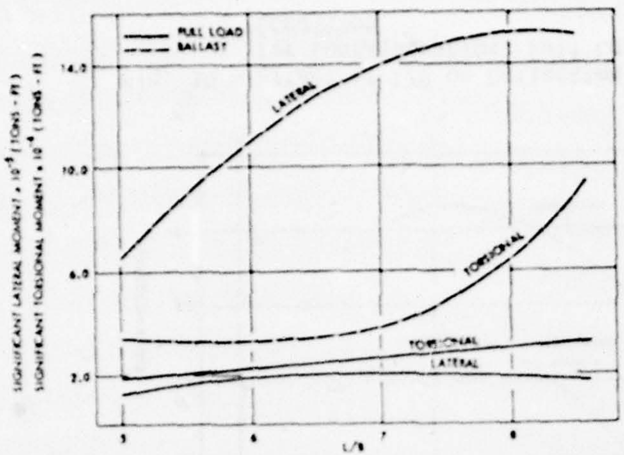

Fig. 24 - Effect of L/B on Torsional and Lateral Bending Moment for Cargo Ships

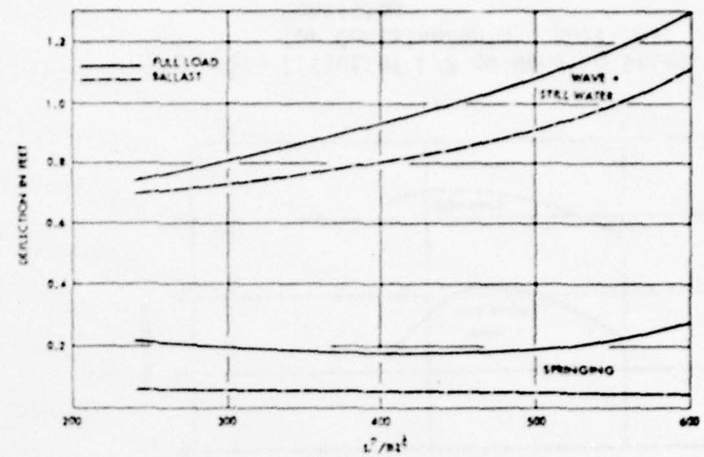

Fig. 25 - Effect of $L^2/BI^{\frac{1}{4}}$ on Deflection for Cargo Ships

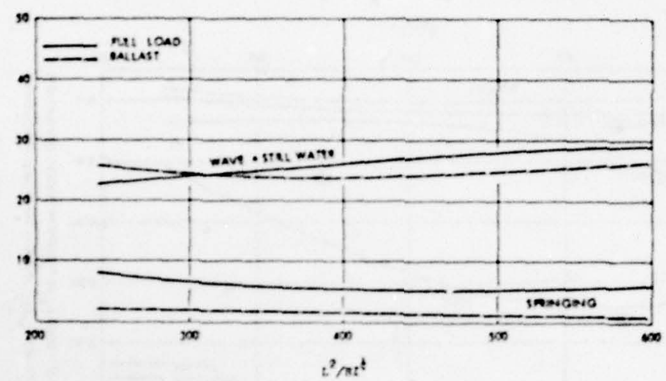

Fig. 26 - Effect of $L^2/BI^{\frac{1}{4}}$ on Vertical Bending Stress for Cargo Ships

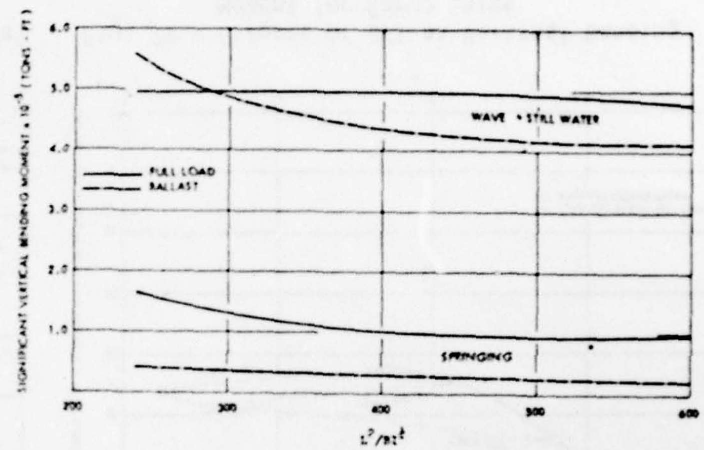

Fig. 27 - Effect of $L^2/BI^{\frac{1}{4}}$ on Vertical Bending Moment for Cargo Ships

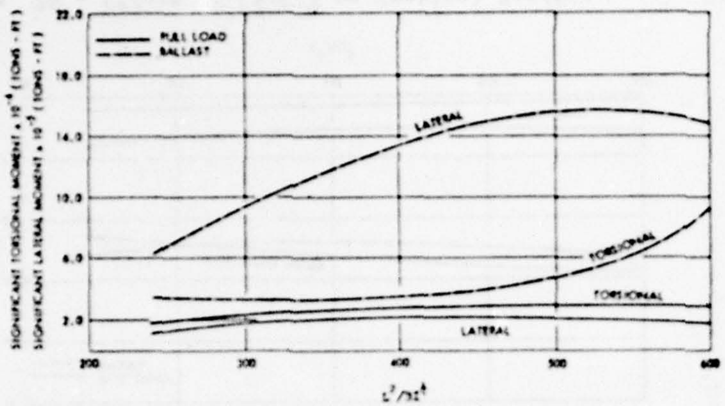

Fig. 28 - Effect of $L^2/BI^{\frac{1}{4}}$ on Torsional and Lateral Moment for Cargo Ships

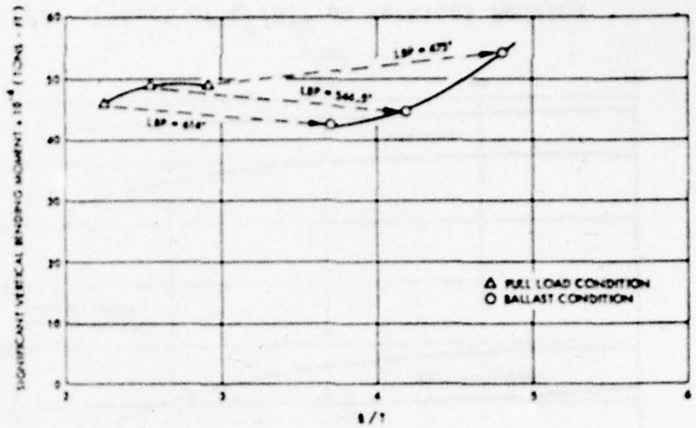

Fig. 29 - Effects of B/T on Vertical Bending Moment for Cargo Ships

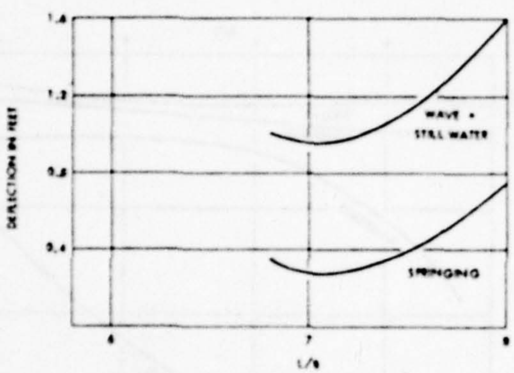

Fig. 30 - Effect of L/B on Deflection for Containerships, Full Load Condition

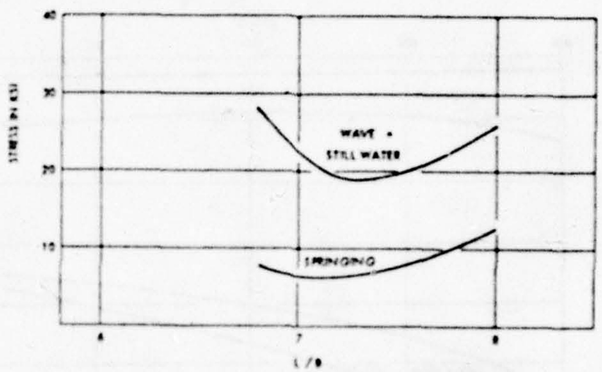

Fig. 31 - Effect of L/B on Bending Stress for Containerships, Full Load Condition

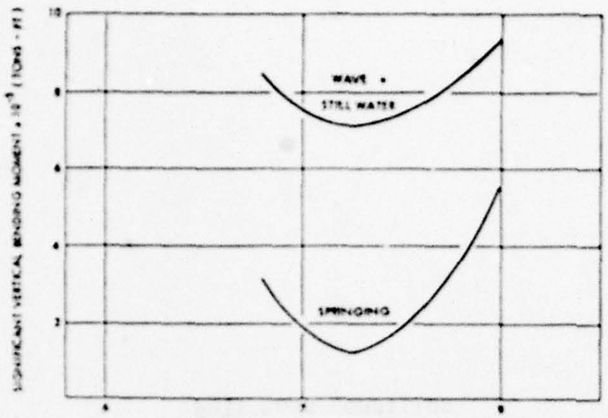

Fig. 32 - Effect of L/B on Vertical Bending Moment for Containerships, Full Load Condition

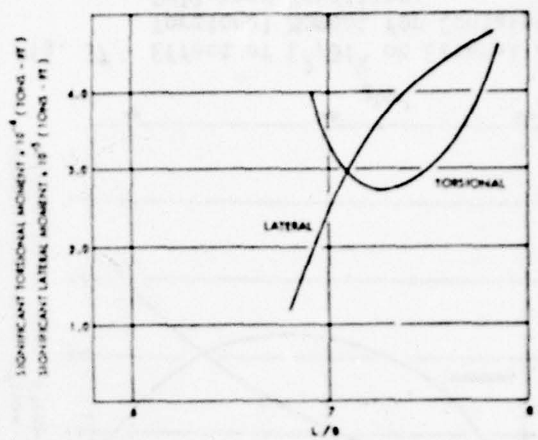

Fig. 33 - Effect of L/B on Torsional and Lateral Moment for Containerships. Full Load Condition

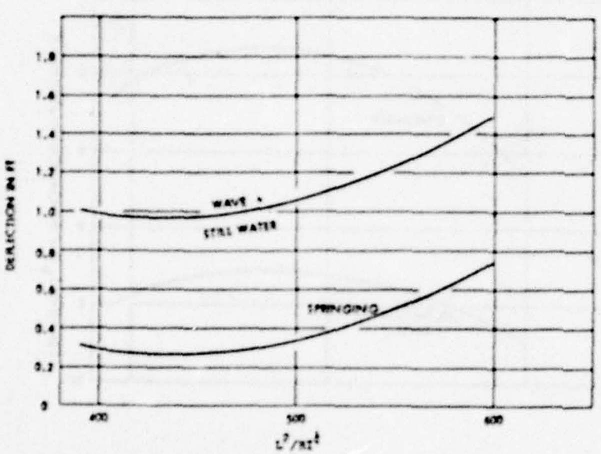

Fig. 34 - Effect of $L^2/BI^{\frac{1}{4}}$ on Deflection for Containerships, Full Load Condition

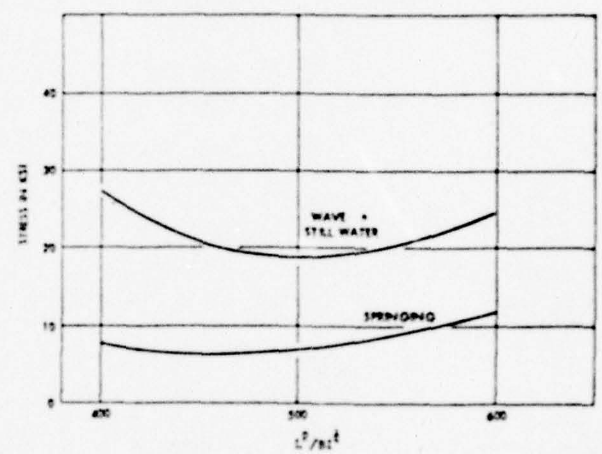

Fig. 35 - Effect of $L^2/BI^{\frac{1}{4}}$ on Bending Stress for Containerships, Full Load Condition

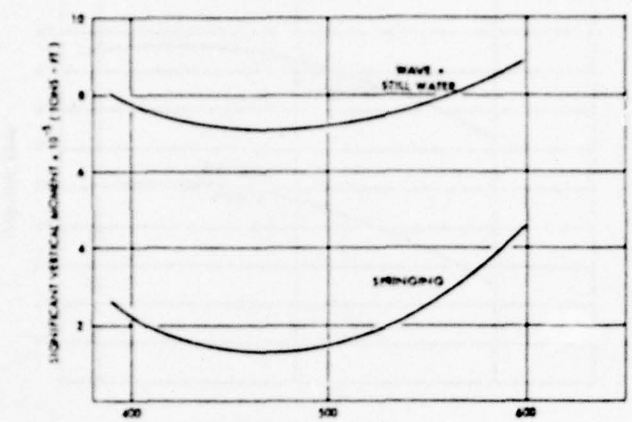

Fig. 36 - Effect of $L^2/BI^{1/4}$ on Vertical Bending Moment for Cortainerships, Full Load Condition

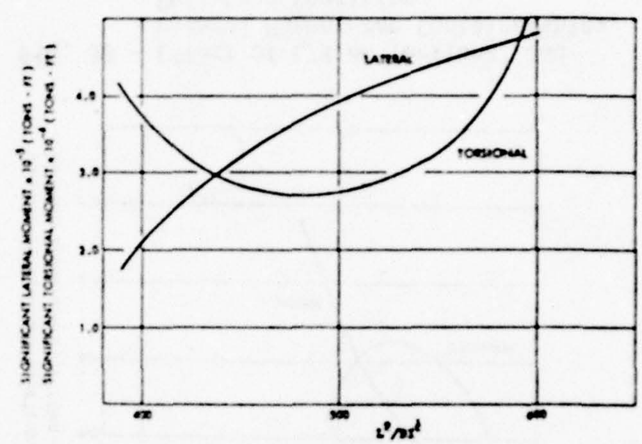

Fig. 37 - Effect of $L^2/BI^{1/4}$ on Lateral and Torsional Moment for Containerships, Full Load Condition

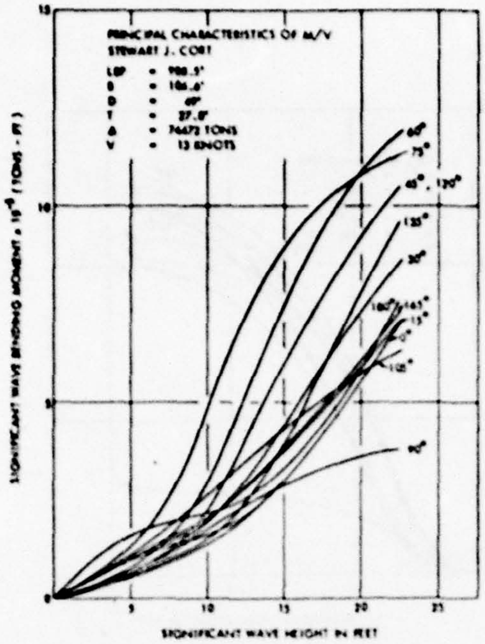

Fig. 38 - Effect of Wave Height and Heading on Sea Loads of Great Lakes Ore Carrier, Full Load Condition

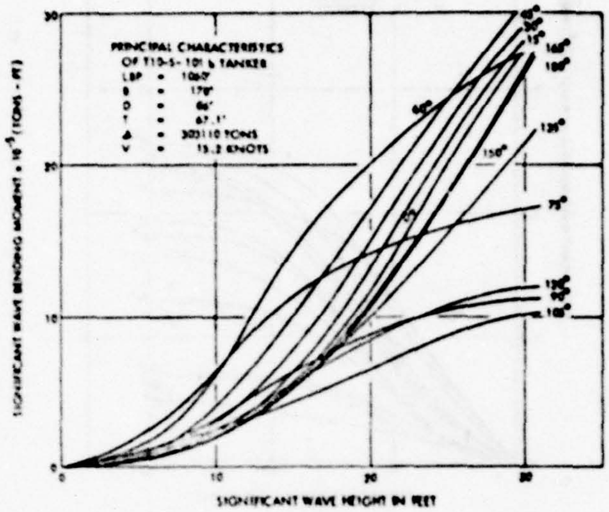

Fig. 39 - Effect of Wave Height and Heading on Sea Loads of Tanker, Full Load Condition

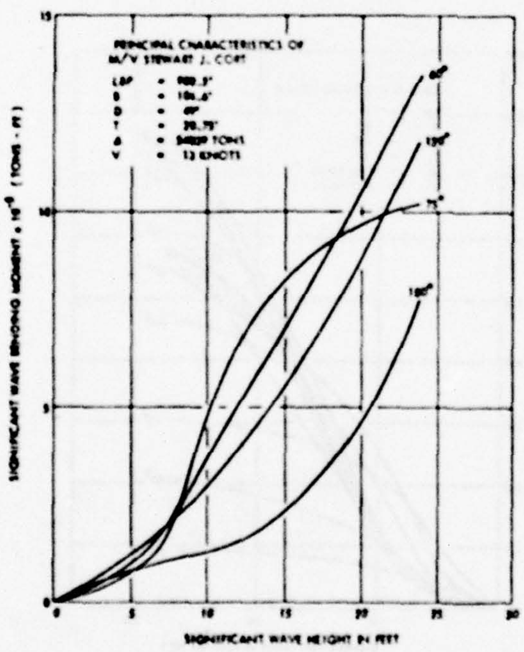

Fig. 40 - Effect of Wave Height and Heading on Sea Loads of Great Lakes Ore Carrier, Ballast Condition

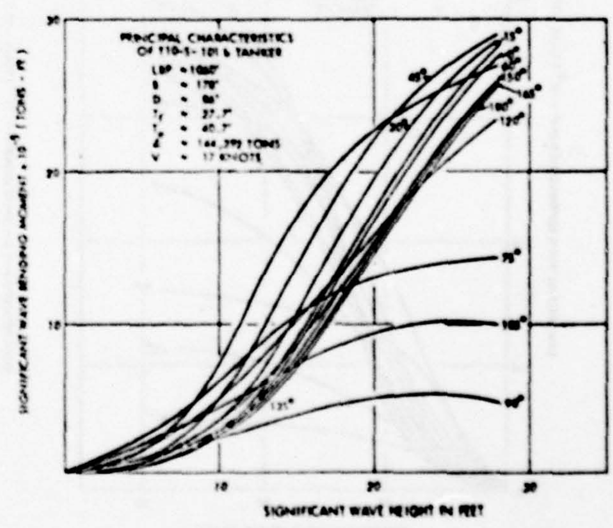

Fig. 41 - Effect of Wave Height and Heading on Sea Loads of Tanker, Ballast Condition

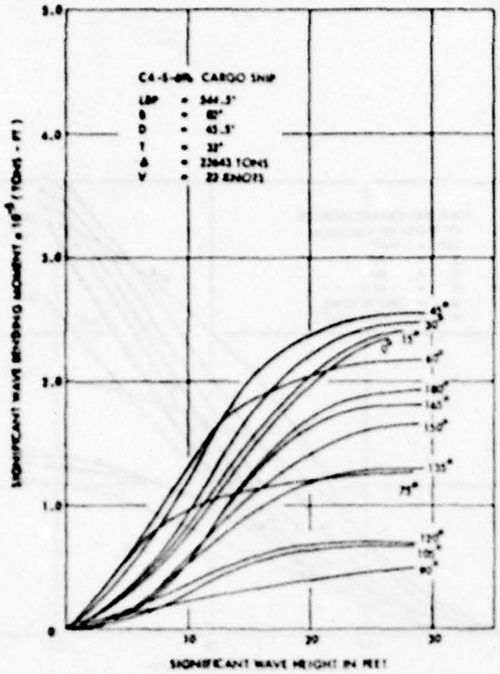

Fig 42 - Effect of Wave Height and Heading on Sea Loads of Cargo Ship, Full Load Condition

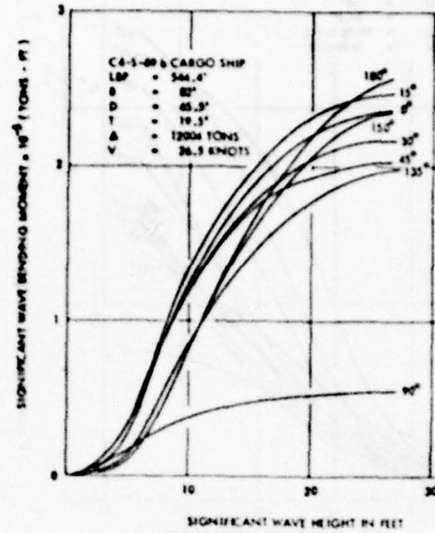

Fig. 43 - Effect of Wave Height and Heading on Sea Loads of Cargo Ship, Ballast Condition

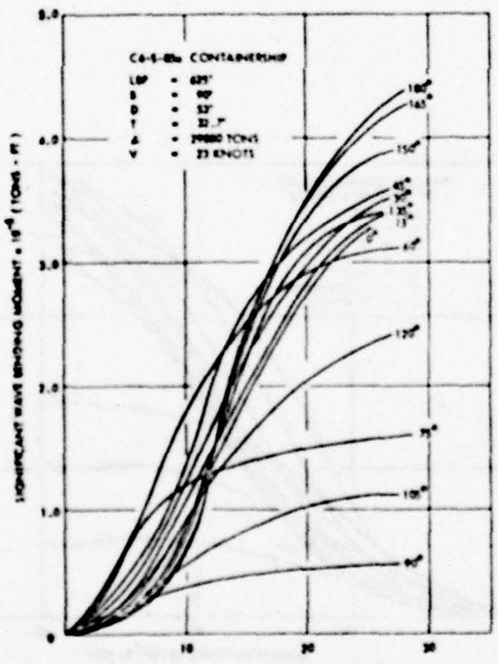

Fig. 44 - Effect of Wave Height and Heading on Sea Loads of C6 Containership, Full Load Condition

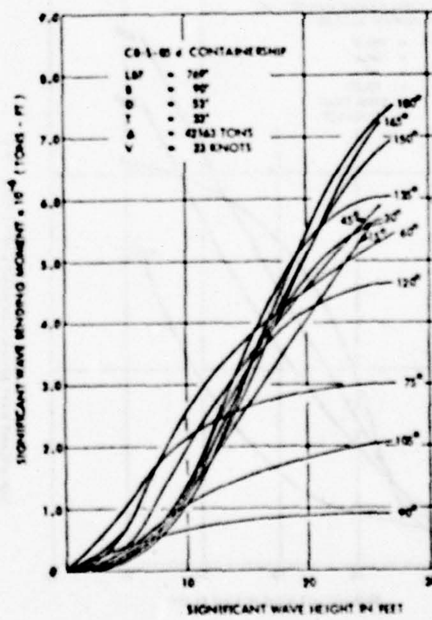

Fig. 45 - Effect of Wave Height and Heading on Sea Loads of C8 Containership, Full Load Condition

and 20.5 feet for Greak Lakes ships. Within these limits of sea states, the wave loads associated with the maximum vertical bending moment were adopted for the vibration analysis.

4.5.2 Wave Loads for Springing Condition - For the springing condition, the two-node frequency of the ship was first calculated. Using the two-node frequency, ω_1, the wave frequencies and headings which could cause springing were determined from the relation:

$$\omega_e = \omega_1 = \omega_w - \frac{\omega_w^2 U}{g} \cos \alpha \tag{11}$$

where α is the heading angle, beginning with zero degrees corresponding to following seas.

Among all the sets of peak energy wave frequencies and heading angles, a set of waves and headings associated with the maximum rigid-hull significant bending moment was determined. The sea loads for this set of headings and waves were adopted for the springing analysis.

4.5.3 Approximate Method for Determining the Flexible Hull Bending Moment - Theoretically the maximum vibration bending moment can only be determined by calculating the bending moment associated with the entire wave spectrum, a task beyond the scope of this project. In view of the many uncertainties in existing vibration theory, an absolute maximum is not of interest. Relative maxima within the accuracy of the existing theory can be obtained by the following approximate method:

Let BM_i, BM_r be the flexible ship bending moment in irregular wave and unit regular waves,

BM_{Ri}, RM_{Rr} be the bending moments in irregular wave and unit regular waves for the same ship assumed to be rigid.

For the same regular wave load, the flexible hull bending moment, BM_r, and the rigid hull bending moment BM_{Rr}, can be calculated. Since the sea load in irregular waves can be regarded as a combination of many regular wave loads, the ratio between the rigid hull bending moment, BM_{Ri}, and the flexible hull bending moment, BM_i, for the same irregular wave load, can be determined approximately as follows:

$$\frac{BM_{Ri}}{BM_i} = \frac{BM_{Rr}}{BM_r} \tag{12}$$

In high-energy waves, the deflection of the flexible hull is small in comparison with the rigid-hull motions. The differences between the seakeeping and flexible hull bending moments are small. For this case, the above equation is quite good. In the springing condition, the deflection of the flexible hull may be much greater

than the rigid-hull motions. In that case, the above equation may induce some errors. It was indicated in previous discussion that the state-of-the-art is inaccurate for the springing condition unless the neglected hydroelastic effects are taken into consideration. In the absence of more accurate methods for analyzing springing, the above equation can be used for estimating the approximate springing moment.

According to Goodman, Reference 6, the two-node mode vibration predominates at and around the two-node natural frequency. If this is true, the errors due to equation (12) should be small even in the springing condition.

Equation (12) was used to calculate the vertical bending moment for the flexible ship for both the wave bending case and the springing case. The bending moment in irregular waves for the rigid ship (BM_{Ri}) was calculated using the MIT5D. This program was also used to calculate BM_{Rr}, the wave bending moment in unit regular waves for the rigid ship. The bending moment for the flexible ship in regular waves, BM_r, was calculated using the modified BEAMRESPONSE program.

For the wave bending case, the significant wave height was taken as 25 ft. for the ocean-going ships and 20.5 ft. for Great Lakes ships. For the springing case the wave height was chosen to correspond to the heading that gave the maximum bending moment for ω_{e_1}, the encounter frequency, equal to ω_1, the natural 2-noded hull frequency; where ω_{e_1} corresponds to the peak frequency of the wave spectrum.

In the curves in Figures 5 through 37, the still-water bending moment was added to the wave bending moment, calculated as described above, to obtain BM_V, but was not added to the plotted springing bending moment.

4.6 Effects of Hull Materials

The effects of hull materials on the hull flexibility were considered in the study in the following manner:

(1) High-Strength Steel

Classification societies usually allow certain reductions in the scantlings of the ship structure if high-strength steel is used. This reduction in scantlings will reduce the moment of inertia of the ship section with a corresponding increase in the hull flexibility. Using the two-node frequency as the parameter for hull flexibility, the increase in flexibility can be determined from the relationship

$$\omega_h = \omega_s \sqrt{\frac{I_h}{I_s}} \tag{13}$$

where I_h and I_s are the moment of inertia of the cross-section with and without high-strength steel.

Accordingly, the effects of the high-strength steel can be accounted for by properly using the value of the moment of inertia.

(2) Aluminum

Since the modulus of elasticity of aluminum is less than that of steel, both the moment of inertia and the modulus of elasticity must be taken into consideration as in the expression

$$\omega_a = \omega_s \sqrt{\frac{I_a E_a}{I_s E_s}} \tag{14}$$

where

ω_a and ω_s are the two-node frequencies for aluminum and steel, respectively;

I_a and I_s are the moments of inertia for aluminum and steel, respectively;

E_a and E_s are the modulii of elasticity for aluminum and steel, respectively.

Thus, the effects of using aluminum and steel can be taken into consideration by evaluating the product of cross-section moment of inertia and the modulus of elasticity.

(3) Composite Materials

Hulls with mild steel and higher strength steels can be readily compared since the modulii of elasticity of these two materials are the same.

For ships constructed of both mild steel and aluminum, the problem is more complicated. For this case, the concept of equivalent moment of inertia must be used. Letting A_{ai}, A_{si} be the cross-sectional areas of the aluminum and steel members; y_{ai}, y_{si} the distance from the center of gravity of those areas to the neutral axis of the ship cross-section, the equivalent moment of inertia is defined as

$$I^* = \sum_{i=1}^{N}\left(A_{si} \cdot y_{si}^{2} + I_{si}\right) + \sum_{i=1}^{M}\frac{E_a}{E_s}\left(A_{ai}y_{ai}^{2} + I_{ai}\right) \qquad (15)$$

where I_{si}, I_{ai} are the moments of inertia of each structural member about its own center of gravity;

N, M are the numbers of the steel and aluminum members in the cross section.

The effect of the aluminum structure is included in the equivalent moment of inertia. As a special case when the entire hull is made of aluminum, Equation (15) reduces to

$$I^* = \sum\frac{E_a}{E_s}\left(A_{ai}y_{ai}^{2} + I_{ai}\right) = \frac{E_a}{E_s}\, I_a \qquad (16)$$

5.0 SELECTION OF REPRESENTATIVE SHIPS FOR ANALYSIS

The following four vessels were selected as vehicles for conducting the hull flexibility study:

(1) Great Lakes ore carrier STEWART J. CORT.
(2) 264,000 dwt U.S. flag tank vessel, designated T10-S-101b.
(3) C6-S-85a and C8-S-85d family of containerships.
(4) C4-S-69b general cargo vessel of MICHIGAN class.

Characteristics of the above vessels, and the proposed parametric variations in dimensions, are considered in the following paragraphs.

Each of the vessels was studied for one full load and one representative ballast condition. Effect of dimensional variations on full load service speed was ignored. For each set of parametric variations of a given parent vessel, one value each of full load and ballast speeds, corresponding to the parent vessel characteristics, was assumed.

As indicated earlier, the required evaluation of the effects of changes in depth and structural materials was obtained by appropriate variation in moment of inertia.

5.1 Great Lakes Ore Carrier STEWART J. CORT

The matrix shown in Table 5 was prepared assuming constant values of breadth, B, and draft, T. These assumptions reflect realistic limits for the foreseeable future, reflecting lock dimensions and operating draft constraints. The 1,000 ft overall length reflects existing maximum permissible length for transit of the Poe Locks. It is understood, however, that this constraint may be relaxed to permit length increases of about 100 ft.

Accordingly, a five-ship parallel body series based on the present CORT, with length increases to 1,300 ft overall and length reductions to 800 ft overall, was investigated. It was assumed that these changes in dimensions would be accomplished by simple addition and subtraction of parallel mid-body, for constant breadth and draft. The full load service speed of the CORT was assumed constant for the series and a higher service speed was assumed for the lighter ballast draft.

Vessels similar to the CORT have been built to the same overall length and breadth constraints, but with increased depth to obtain the higher cubic capacity required for coal transport. The most recent vessel built for this service is the BELLE RIVER, Bay Shipbuilding Hull No. 716, with D = 56 ft. Accordingly, the series includes two values of depth, with D = 49 ft for the shorter vessels and D = 56 ft for the longer vessels. The 1,000 ft

TABLE 5

PROPOSED VARIATION IN DIMENSIONS OF

GREAT LAKES VESSEL "STEWART J. CORT"

Length, overall, ft.	800	900	1000 (Basic Design)	1200	1300
Length, B.P., ft., L	788.5	888.5	988.5	1188.5	1288.5
Breadth, mld., ft., B			← 104.60 →		
Depth, mld., ft., D			← 49.0		
	-	-	56.0 →		
Draft, full load, keel, ft., T			← 27.83 →		
Displacement, mld., f.w., l. tons	57,834	65,917	74,000	90,166	98,249
C_B	0.907	0.918	0.926	0.939	0.943
$L/_B$	7.538	8.494	9.450	11.362	12.318
$L/_D$ for D = 49.0	16.092	18.133	20.173	24.255	-
for D = 56.0	-	-	17.652	21.223	23.009
$B/_D$ for D = 49.0	2.133	2.133	2.133	-	-
for D = 56.0	-	-	1.866	1.866	1.866
$B/_T$			← 3.758 →		

TABLE 6 - PROPERTIES OF GREAT LAKES ORE CARRIER "STEWART J. CORT"
FULL LOAD CONDITION

LBP = 988.5'
B = 104.6'
D = 49'
T = 27.83'
Δ = 74472 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft^2)	MO. INERTIA (ft^4)	MASS DENSITY (ton-sec^2/ft^2)
1	1.080000E+02	1.928600E+06	1.000000E+04	1.100000E+00
2	9.600000E+01	1.928600E+06	1.668000E+04	6.220000E+00
3	9.600000E+01	1.928600E+06	1.668000E+04	6.220000E+00
4	1.040000E+02	1.928600E+06	1.668000E+04	6.220000E+00
5	9.600000E+01	1.928600E+06	1.668000E+04	6.220000E+00
6	1.040000E+02	1.928600E+06	1.668000E+04	6.220000E+00
7	9.600000E+01	1.928600E+06	1.668000E+04	6.220000E+00
8	9.600000E+01	1.928600E+06	1.668000E+04	6.220000E+00
9	1.190000E+02	1.928600E+06	1.668000E+04	6.220000E+00
10	8.500000E+01	1.928600E+06	6.950000E+03	3.500000E+00

SECTION	ELAS. FDN (ton/ft^2)	ROT. ELAS. FDN	SHEAR AREA (ft^2)	POISSON RATIO
1	1.100000E+00	0.	7.000000E+00	3.000000E-01
2	2.850000E+00	0.	1.400000E+01	3.000000E-01
3	2.850000E+00	0.	1.400000E+01	3.000000E-01
4	2.850000E+00	0.	1.400000E+01	3.000000E-01
5	2.850000E+00	0.	1.400000E+01	3.000000E-01
6	2.850000E+00	0.	1.400000E+01	3.000000E-01
7	2.850000E+00	0.	1.400000E+01	3.000000E-01
8	2.850000E+00	0.	1.400000E+01	3.000000E-01
9	2.850000E+00	0.	1.400000E+01	3.000000E-01
10	2.200000E+00	0.	6.000000E+00	3.000000E-01

TABLE 7 - PROPERTIES OF GREAT LAKES ORE CARRIER "STEWART J. CORT"
BALLAST CONDITION

LBP = 988.5'
B = 104.6'
D = 49'
T = 20.75'
Δ = 54840 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft²)	MO. INERTIA (ft⁴)	MASS DENSITY (ton-sec²/ft²)
1	1.080000E+02	1.928600E+06	5.000000E+03	1.100000E+00
2	9.600000E+01	1.928600E+06	1.668000E+04	5.800000E+00
3	9.600000E+01	1.928600E+06	1.668000E+04	5.800000E+00
4	1.040000E+02	1.928600E+06	1.668000E+04	5.800000E+00
5	9.600000E+01	1.928600E+06	1.668000E+04	5.910000E+00
6	1.040000E+02	1.928600E+06	1.668000E+04	5.910000E+00
7	9.600000E+01	1.928600E+06	1.668000E+04	5.910000E+00
8	9.600000E+01	1.928600E+06	1.668000E+04	5.910000E+00
9	1.190000E+02	1.928600E+06	1.668000E+04	5.910000E+00
10	8.500000E+01	1.928600E+06	6.950000E+03	3.500000E+00

SECTION	ELAS. FDN (ton/ft²)	ROT. ELAS. FDN	SHEAR AREA (ft²)	POISSON RATIO
1	1.100000E+00	0.	7.000000E+00	3.000000E-01
2	2.850000E+00	0.	1.400000E+01	3.000000E-01
3	2.850000E+00	0.	1.400000E+01	3.000000E-01
4	2.850000E+00	0.	1.400000E+01	3.000000E-01
5	2.850000E+00	0.	1.400000E+01	3.000000E-01
6	2.850000E+00	0.	1.400000E+01	3.000000E-01
7	2.850000E+00	0.	1.400000E+01	3.000000E-01
8	2.850000E+00	0.	1.400000E+01	3.000000E-01
9	2.850000E+00	0.	1.400000E+01	3.000000E-01
10	2.200000E+00	0.	6.000000E+00	3.000000E-01

TABLE 8 - VARIATION OF PROPORTIONS AND RESPONSES OF GREAT LAKES ORE CARRIERS - FULL LOAD CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \times 10^{-3}$ (ton-ft)	$BM_L \times 10^{-3}$ (ton-ft)	$BM_T \times 10^{-3}$ (ton-ft)	Deflta (ft)	L/B	$\frac{L^3}{BI^3}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
60	788.5	3.7639	57,486	16,680	894	463	45.6	1.2275	7.5274	522.27	3.0588	21.441	21,726
180					429			0.5989					10,425
60	888.5	3.7639	66,950	16,680	934	570	40.0	1.7210	8.4943	644.10	2.5940		21,480
180					550			0.7980					13,068
75	988.5	3.7630	74,472	16,680	1049	696	41.6	2.1894	9.4499	821.97	2.1077		25,502
180					799			1.6923					19,422
75	1188.5	3.7636	85,510	25,950	1550	1030	70.4	3.3450	11.3623	1063.98	1.5230		24,220
180					1840			5.4902					28,752
75	1288.5	3.7649	90,267	33,360	2099	1206	95.4	3.4020	12.3007	1172.7	1.4252		29,150
180					2844			8.8349					39,508

TABLE 9 - VARIATION OF PROPORTIONS AND RESPONSES OF GREAT LAKES ORE CARRIERS - BALLAST CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \times 10^{-3}$ (ton-ft)	$BM_L \times 10^{-3}$ (ton-ft)	$BM_T \times 10^{-3}$ (ton-ft)	Deflta (ft)	L/B	$\frac{L^3}{BI^3}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
60	788.5	4.9788	45,645	16,680	707	346	34.0	0.9596	7.5274	522.27	3.1412	21.939	17,241
180					127			0.1744					3,087
60	888.5	5.0424	49,850	16,680	860	394	38.9	1.7200	8.4943	664.10	2.6400		20,970
180					220			0.2950					5,364
60	988.5	5.0424	54,839	16,680	1087	483	45.6	2.2879	9.4499	821.97	2.1495		26,419
180					307			0.6963					7,456
60	1188.5	4.9950	67,100	25,950	1640	789	66.5	3.195	11.3623	1063.98	1.8210		25,620
180					840			2.450					13,122
60	1288.5	4.9880	74,549	33,360	2062	1042	83.3	3.3169	12,3007	1172.7	1.9263		28,641
180					1260			3.9141					17,620

parent was studied for both values of depth, thus providing for a six-ship series.

Characteristics of the proposed series at full-load draft are summarized in Table 5. The range of values of L/B indicate proportions that extend well beyond current practice on the Great Lakes. The values of L/D , from 16 to over 24, exceed oceangoing limits and extend from current Great Lakes limiting values of 20 to well beyond current design practice. Values of B/T are constant and reflect the existing values of this ratio for vessels designed to transit the Poe Locks.

Sectional properties used in the analysis are summarized in Tables 6 and 7 and assumed speeds are included with the response data, Tables 8 and 9.

5.2 <u>264,000 DWT Tanker</u>

The proposed matrix of systematic dimensional variations for the T10-S-101b tanker is shown in Table 10. Draft and displacement values are specific to the full-load condition.

The matrix was prepared assuming constant values of displacement and draft. A systematic variation in the ratio L/B, was assumed, thus providing for corresponding variations in the significant ratios L/D, B/D. This approach differs from the alternatives normally examined by the ship designer in that the owner's requirements generally include defined values of deadweight and draft restriction. Light ship and deadweight values will vary with proportions. However the constant displacement series is a reasonable approximation to the designer's constant deadweight approach and it provides a practical basis for study analysis.

The five design points indicated in the table were further examined by analysis with the HYDRONAUTICS' concept design computer program, to obtain realistic values of depth for the assumed variations in hull dimensions and proportions. This computer analysis also provided the necessary weight information to support preparation of a systematic variation of weight curves for the four variations of the parent.

The computer design analysis indicated that required depth variations would be small, approximately 1.25 ft from the base value of 86 ft. Further, for a constant service speed and C_B, power requirements varied only approximately 800 hp. Accordingly, it was considered reasonable to hold the original values of D and C_B constant for basic variation in parameters.

TABLE 10

PROPOSED VARIATIONS IN DIMENSIONS

OF 264,000 DWT TANKER

Item	Nominal $L/_B$ Value				
	5	5.5	6 (Basic Design)	6.5	7
Length, B.P., ft., L	968.00	1016.00	1060.00	1105.00	1147.00
Breadth, mld., ft., B	194.91	185.70	178.00	170.75	164.50
Depth, mld., ft., D			86.00		
	-	-	* 76.00		
Draft, full load, mld, ft.,T			67.0625		
Displacement,mld., l. tons			304573		
C_B	0.8425	0.8425	0.84247	0.8425	0.8425
$L/_B$	4.9664	5.4712	5.9551	6.4714	6.9726
$L/_D$ for D = 86.00	11.2558	11.8140	12.3256	12.8488	13.3372
for D = 76.00	-	-	13.9474	14.5395	15.0921
$B/_D$ for D = 86.00	2.2664	2.1593	2.0698	1.9855	1.9128
for D = 76.00	-	-	2.3421	2.2467	2.1645
$B/_T$	2.9064	2.7691	2.6542	2.5461	2.4529

*Value of 76 ft. is inadequate to obtain freeboard draft of 67 ft., hence this series is of academic interest only.

To obtain ship characteristics with high values of the ratio L/D, a reduced value of D = 76 ft was arbitrarily assumed for the three longest designs. This three-point series is academic in that the 76 ft depth is inadequate to obtain the freeboard draft of 67 ft.

Sectional properties used in the analysis are summarized in Tables 11 and 12 and assumed speeds are included in the response data, Tables 13 and 14.

5.3 "C4" General Cargo Vessel

An existing U.S. flag general cargo vessel, designated C4-S-69b, was selected for study. The MICHIGAN of this class has been the subject of earlier studies under Ship Sturcture Committee sponsorship. The proposed matrix of systematic dimensional variations is shown in Table 15. Draft and displacement values are specific to the full load condition.

The matrix was prepared assuming constant values of displacement, draft, and block coefficient, C_B. A systematic variation in the ratio L/B was assumed, thus providing for corresponding variations in the significant ratios L/D and B/D. It is recognized that the two longest vessels, with highest values of L/B and lowest values of B/D, may have marginal stability characteristics. However, the series was retained for study since the intent was to investigate systematically the effect of varying ship proportions.

Sectional properties used in the analysis are summarized in Tables 16 and 17 and assumed speeds are included in the response data, Tables 18 and 19.

5.4 "C6" and "C8" Containerships

Existing C6-S-85a and C8-S-85d containerships were selected for the study. The original "C6" design was completed in 1968 and has been in service since that time for two U.S. flag operators. Recently, a group of the original "C6" vessels was converted by lengthening 144 ft and, subsequently, new construction of this latter configuration, designated C8-S-85d, was initiated and is currently under construction. Design and characteristics data for both configurations is currently available. Further, operational data exists for the "C6" design and will become available for the longer "C8" design. Accordingly, selection of these vessels for study provided points in a ship series wherein analytical studies can be related to actual design and operating experience.

TABLE 11 PROPERTIES OF 264,000 DWT "T10" TANKER,
FULL LOAD CONDITION

LBP = 1060'
B = 178'
D = 86'
T = 67.1'
Δ = 304573 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft²)	MO. INERTIA (ft⁴)	MASS DENSITY (ton-sec²/ft²)
1	1.000000E+02	1.928600E+06	9.750000E+03	7.851400E+00
2	1.000000E+02	1.928600E+06	6.398700E+04	1.871710E+01
3	1.000000E+02	1.928600E+06	1.182230E+05	2.026320E+01
4	1.000000E+02	1.928600E+06	1.182230E+05	2.026320E+01
5	1.000000E+02	1.928600E+06	1.182230E+05	1.638120E+01
6	1.000000E+02	1.928600E+06	1.182230E+05	2.110250E+01
7	1.000000E+02	1.928600E+06	1.182230E+05	2.110250E+01
8	1.000000E+02	1.928600E+06	1.182230E+05	2.098130E+01
9	1.000000E+02	1.928600E+06	1.182230E+05	2.004550E+01
10	1.000000E+02	1.928600E+06	6.449200E+04	1.271970E+01
11	1.000000E+02	1.928600E+06	1.076000E+04	3.906500E+00

SECTION	ELAS. FDN (ton/ft²)	ROT. ELAS. FDN	SHEAR AREA (ft²)	POISSON RATIO
1	2.650000E+00	0.	7.299600E+00	3.000000E-01
2	4.850000E+00	0.	1.064100E+01	3.000000E-01
3	5.086000E+00	0.	1.398200E+01	3.000000E-01
4	5.086000E+00	0.	1.398200E+01	3.000000E-01
5	5.086000E+00	0.	1.398200E+01	3.000000E-01
6	5.086000E+00	0.	1.398200E+01	3.000000E-01
7	5.086000E+00	0.	1.398200E+01	3.000000E-01
8	5.086000E+00	0.	1.398200E+01	3.000000E-01
9	5.086000E+00	0.	1.398200E+01	3.000000E-01
10	4.437000E+00	0.	9.600000E+00	3.000000E-01
11	2.097000E+00	0.	5.218000E+00	3.000000E-01

TABLE 12 PROPERTIES OF 264,000 DWT "T10" TANKER, BALLAST CONDITION

LBP = 1060'
B = 178'
D = 86'
T = 35.925'
Δ = 144392 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft^2)	MO. INERTIA (ft^4)	MASS DENSITY (ton-sec^2/ft^2)
1	1.000000E+02	1.928570E+06	9.750000E+03	4.651000E+00
2	1.000000E+02	1.928570E+06	6.393700E+04	1.213400E+01
3	1.000000E+02	1.928570E+06	1.182230E+05	1.657900E+01
4	1.000000E+02	1.928570E+06	1.182230E+05	2.038600E+01
5	1.000000E+02	1.928570E+06	1.182230E+05	1.390000E+01
6	1.000000E+02	1.928570E+06	1.182230E+05	1.887900E+01
7	1.000000E+02	1.928570E+06	1.182230E+05	2.043800E+01
8	1.000000E+02	1.928570E+06	1.182230E+05	1.843100E+01
9	1.000000E+02	1.928570E+06	1.182230E+05	1.468900E+01
10	1.000000E+02	1.928570E+06	6.449200E+04	6.321000E+00
11	1.000000E+02	1.928570E+06	1.076000E+04	1.051000E+00

SECTION	ELAS. FDN (ton/ft^2)	ROT. ELAS. FDN	SHEAR AREA (ft^2)	POISSON RATIO
1	2.360000E+00	0.	7.300000E+00	3.000000E-01
2	4.800000E+00	0.	1.064100E+01	3.000000E-01
3	5.036000E+00	0.	1.398200E+01	3.000000E-01
4	5.036000E+00	0.	1.398200E+01	3.000000E-01
5	5.036000E+00	0.	1.398200E+01	3.000000E-01
6	5.036000E+00	0.	1.398200E+01	3.000000E-01
7	5.036000E+00	0.	1.398200E+01	3.000000E-01
8	5.036000E+00	0.	1.398200E+01	3.000000E-01
9	5.000000E+00	0.	1.398200E+01	3.000000E-01
10	3.543000E+00	0.	9.600000E+00	3.000000E-01
11	5.140000E-01	0.	5.21[illegible]E+00	3.000000E-01

TABLE 13 - VARIATION OF PROPORTIONS AND RESPONSES OF TANK VESSELS - FULL LOAD CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \times 10^{-3}$ (ton-ft)	$BM_L \times 10^{-3}$ (ton-ft)	$BM_T \times 10^{-4}$ (ton-ft)	Defltn (ft)	L/B	$\frac{L^3}{BI^4}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
50	968	2.905	302,850	129,450	4404	1659	127.5	1.4855	4.9664	252.44	2.537	25.6374	22,839
180					1721			0.6753					
50	1016	2.769	303,010	123,010	4750	1850	135.0	1.1030	5.4712	296.82	2.302	"	26,500
180					1102			0.4520					
60	1060	2.653	303,110	118,220	4993	2055	130.2	1.0518	5.9550	340.42	2.347	"	28,355
180					844			0.3607					
50	1105	2.546	303,120	113,200	5042	2250	120.0	1.4490	6.4714	389.89	2.050	"	29,200
180					900			0.3940					
50	1147	2.451	304,020	109,260	5053	2459	108.2	2.7092	6.9726	439.89	1.9773	"	30,937
180								0.6516					

TABLE 14 - VARIATION OF PROPORTIONS AND RESPONSES OF TANK VESSELS - BALLAST CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \times 10^{-3}$ (ton-ft)	$BM_L \times 10^{-4}$ (ton-ft)	$BM_T \times 10^{-3}$ (ton-ft)	Defltn (ft)	L/B	$\frac{L^3}{BI^4}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
45	969	5.699	143,870	129,450	4827	603	185.7	1.6482	4.9664	252.44	3.231	28.712	25,033
180					2258			0.7704					11,714
45	1016	5.430	144,100	123,010	4920	802	195.0	1.8600	5.4712	296.82	2.7602		26,300
180					1995			0.7490					10,064
45	1060	5.205	144,392	118,220	5054	933	192.6	2.189	5.9550	340.42	2.8953		28,701
180					1814			0.7863					10,310
45	1105	4.993	144,200	113,200	5230	950	169.0	2.5700	6.4714	389.89	2.802		31,462
180					2010			0.9360					11,650
50	1147	4.810	143,859	109,260	5477	916	137.9	3.1490	6.9726	439.89	2.582		33,532
180					2436			1.3462					14,916

TABLE 15

PROPOSED VARIATIONS IN DIMENSIONS OF C4-S-69b

GENERAL CARGO VESSEL

Item	Nominal $L/_B$				
	5.1	5.8	6.6 (Basic Design)	7.5	8.5
Length, B.P., ft., L.	475	509	544	579	614
Breadth, mld., ft., B.	94	87.5	82	77	72.5
Depth, mld., ft., D.			45.6		
Draft, freeboard, mld, ft., T.			32.0		
Displacement, mld., l-tons			22500		
C_B	0.5512	0.5526	0.5517	0.5520	0.5528
$L/_B$	5.0532	5.8171	6.6341	7.5195	8.4690
$L/_D$ for D = 45.5	10.4396	11.1868	11.9560	12.7253	13.4945
$B/_D$ for D = 45.5	2.0659	1.9231	1.8022	1.6923	1.5934
$B/_T$	3.0519	2.8409	2.6623	2.5000	2.3539

TABLE 16 - PROPERTIES OF C4-S-69b GENERAL CARGO VESSEL
FULL LOAD CONDITION

LBP = 544'
B = 82'
D = 45.6'
T = 32'
Δ = 22643 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft²)	MO. INERTIA (ft⁴)	MASS DENSITY (ton-sec²/ft⁴)
1	6.850000E+01	1.928600E+06	3.633000E+03	3.877000E-01
2	5.400000E+01	1.928600E+06	5.867000E+03	7.589000E-01
3	5.400000E+01	1.928600E+06	7.000000E+03	1.477100E+00
4	5.400000E+01	1.928600E+06	6.933000E+03	2.513200E+00
5	5.400000E+01	1.928600E+06	7.333000E+03	3.243500E+00
6	5.400000E+01	1.928600E+06	6.917000E+03	3.424300E+00
7	5.400000E+01	1.928600E+06	7.033000E+03	3.160900E+00
8	5.400000E+01	1.928600E+06	6.466000E+03	1.817600E+00
9	5.400000E+01	1.928600E+06	4.333000E+03	1.074400E+00
10	7.400000E+01	1.928600E+06	2.800000E+03	6.575000E-01

SECTION	ELAS. FDN (ton/ft²)	ROT. ELAS. FDN	SHEAR AREA (ft²)	POISSON RATIO
1	0.	0.	4.300000E+00	3.000000E-01
2	5.520000E-01	0.	5.970000E+00	3.000000E-01
3	1.215000E+00	0.	6.950000E+00	3.000000E-01
4	1.885000E+00	0.	7.350000E+00	3.000000E-01
5	2.293000E+00	0.	7.070000E+00	3.000000E-01
6	2.343000E+00	0.	6.460000E+00	3.000000E-01
7	2.275000E+00	0.	6.640000E+00	3.000000E-01
8	1.909000E+00	0.	7.550000E+00	3.000000E-01
9	1.175000E+00	0.	7.110000E+00	3.000000E-01
10	2.230000E-01	0.	4.740000E+00	3.000000E-01

TABLE 17 - PROPERTIES OF C4-S-69b CARGO VESSEL
BALLAST CONDITION

LBP = 544'
B = 82'
D = 45.5'
T = 19.5'
Δ = 12004 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft^2)	MO. INERTIA (ft^4)	MASS DENSITY (ton-sec^2/ft^2)
1	6.850000E+01	1.928600E+06	3.633000E+03	3.877000E-01
2	5.400000E+01	1.928600E+06	5.867000E+03	7.589000E-01
3	5.400000E+01	1.928600E+06	7.000000E+03	1.477100E+00
4	5.400000E+01	1.928600E+06	6.933000E+03	2.513200E+00
5	5.400000E+01	1.928600E+06	7.333000E+03	3.243500E+00
6	5.400000E+01	1.928600E+06	6.917000E+03	3.424300E+00
7	5.400000E+01	1.928600E+06	7.033000E+03	3.160900E+00
8	5.400000E+01	1.928600E+06	6.466000E+03	1.817600E+00
9	5.400000E+01	1.928600E+06	4.333000E+03	1.074400E+00
10	7.400000E+01	1.928600E+06	2.800000E+03	6.575000E-01

SECTION	ELAS. FDN (ton/ft^2)	ROT. ELAS. FDN	SHEAR AREA (ft^2)	POISSON RATIO
1	0.	0.	4.300000E+00	3.000000E-01
2	5.520000E-01	0.	5.970000E+00	3.000000E-01
3	1.215000E+00	0.	6.950000E+00	3.000000E-01
4	1.885000E+00	0.	7.350000E+00	3.000000E-01
5	2.293000E+00	0.	7.070000E+00	3.000000E-01
6	2.343000E+00	0.	6.460000E+00	3.000000E-01
7	2.275000E+00	0.	6.640000E+00	3.000000E-01
8	1.909000E+00	0.	7.550000E+00	3.000000E-01
9	1.175000E+00	0.	7.110000E+00	3.000000E-01
10	2.230000E-01	0.	4.740000E+00	3.000000E-01

TABLE 18 - VARIATION OF PROPORTIONS AND RESPONSES OF GENERAL CARGO VESSELS - FULL LOAD CONDITIONS

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \cdot 10^{-3}$ (ton-ft)	$BM_L \cdot 10^{-3}$ (ton-ft)	$BM_T \cdot 10^{-3}$ (ton-ft)	Defltn (ft)	L/B	$\frac{L^3}{BI^4}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
30	475	2.937	22,641	8406	488	133	18	0.6763	5.0532	250.6	6.767	38.846	22,641
180					169			0.2181					
30	509	2.734	22,642	7850	495	181	21	0.7810	5.8171	314.5	6.312		22,642
180					124			0.1980					
45	544.5	2.563	22,643	7400	489	203	23	0.9196	6.6400	389.8	5.790		22,643
180					103			0.1829					
45	579	2.406	22,614	6850	485	198	27	1.0621	7.5195	478.5	5.293		22,614
180					99			0.1895					
45	614	2.266	22,585	6433	458	176	30	1.2341	8.4689	590.6	4.7212		22,585
180					99			0.2521					

TABLE 19 - VARIATION OF PROPORTIONS AND RESPONSES OF GENERAL CARGO VESSELS - BALLAST CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \cdot 10^{-3}$ (ton-ft)	$BM_L \cdot 10^{-3}$ (ton-ft)	$BM_T \cdot 10^{-3}$ (ton-ft)	Defltn (ft)	L/B	$\frac{L^3}{BI^4}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
15	475	4.825	12,003	8406	542	689	34.6	0.7068	5.0532	250.6	9.1986	44.757	25,572
180					46			0.0601					2,191
30	509	4.491	12,003	7850	481	1025	33.4	0.7352	5.8171	314.5	8.0602		24,503
180					30			0.520					1,650
180	544.5	4.209	12,004	7400	448	1317	34.2	0.7925	6.6400	389.8	7.3272		24,049
180					29			0.0538					1,548
180	579	3.952	12,003	6850	432	1480	48.6	0.8960	7.5195	478.5	6.7435		24,500
180					20			0.0904					1,050
180	614	3.722	12,003	6433	426	1516	85.8	1.0810	8.4689	590.6	6.5350		26,293
180					19.5			0.0403					983

The matrix of ship characteristics selected for study is shown in Table 20. The matrix was prepared assuming constant values of full load draft and depth. The existing "C6" and "C8" parallel mid-body series vessels were selected for the first two design points. The third design point is a further parallel mid-body extension to a length of 875 ft. The resulting characteristics of this design point are currently of academic interest in that the value of L/D = 16.5 exceeds classification society limits.

A fourth design point was obtained by increasing the breadth of the "C8" design to 106 ft, corresponding to the nominal addition of two rows of 8 ft width containers. The 106 ft breadth also corresponds to existing Panama Canal constraints.

Containerships tend to operate with cargo aboard in both outbound and return voyages and draft is near constant for the operating conditions of interest. This conclusion has been verified through discussion with the operators of the "C6" and "C8" vessels. Accordingly, draft and service speed were held constant for the series.

Sectional properties used in the analysis are summarized in Tables 21 and 22 and assumed speed is included in the response data, Table 23.

TABLE 20

PROPOSED VARIATIONS IN DIMENSIONS OF "C6" AND "C8" FAMILY OF CONTAINERSHIPS

Length, B.P., ft., L	625	769	769	875
Breadth, mld., ft., B	90	90	106	90
Depth, mld., ft., D	53	53	53	53
Draft, scantling, mld., ft., T	33	33	33	33
Displacement, mld., l. tons	30300	42100	49585	50770
Block coefficient, C_B	0.5713	0.6452	0.6452	0.6838
$L/_B$	6.9444	8.54444	7.2547	9.7222
$L/_D$	11.7925	14.5094	14.5094	16.5094
$B/_T$	2.7272	2.7272	3.2121	2.7272
$B/_D$	1.6981	1.6981	2.0000	2.0000
Notes	(1)	(2)		

Notes: (1) Basic C6-S-85a design.

(2) Lengthened "C6" design, currently under construction, designated C8-S-85d.

TABLE 21 -PROPERTIES OF C6-S-85a CONTAINERSHIP
FULL LOAD CONDITION

LBP = 625'
B = 90'
D = 53'
T = 32.7'
Δ = 29880 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft²)	MO. INERTIA (ft⁴)	MASS DENSITY (ton-sec²/ft²)
1	8.250000E+01	1.928600E+06	5.083000E+03	7.907000E-01
2	6.250000E+01	1.928600E+06	7.575000E+03	1.802600E+00
3	6.250000E+01	1.928600E+06	1.165800E+04	3.357100E+00
4	6.250000E+01	1.928600E+06	1.316700E+04	4.529000E+00
5	6.250000E+01	1.928600E+06	1.366700E+04	5.061000E+00
6	6.250000E+01	1.928600E+06	1.375000E+04	5.210900E+00
7	6.250000E+01	1.928600E+06	1.425000E+04	4.705800E+00
8	6.250000E+01	1.928600E+06	2.658300E+04	3.490200E+00
9	6.250000E+01	1.928600E+06	1.775000E+04	2.057000E+00
10	3.050000E+01	1.928600E+06	5.750000E+03	1.239600E+00

SECTION	ELAS. FDN (ton/ft³)	ROT. ELAS. FDN	SHEAR AREA (ft²)	POISSON RATIO
1	5.550000E-01	0.	4.000000E+00	3.000000E-01
2	1.415000E+00	0.	6.800000E+00	3.000000E-01
3	2.171000E+00	0.	8.400000E+00	3.000000E-01
4	2.513000E+00	0.	8.400000E+00	3.000000E-01
5	2.513000E+00	0.	8.400000E+00	3.000000E-01
6	2.513000E+00	0.	8.400000E+00	3.000000E-01
7	2.513000E+00	0.	8.400000E+00	3.000000E-01
8	2.327000E+00	0.	7.270000E+00	3.000000E-01
9	1.702000E+00	0.	4.330000E+00	3.000000E-01
10	9.420000E-01	0.	3.000000E+00	3.000000E-01

TABLE 22 - PROPERTIES OF C8-S-85d CONTAINERSHIP
FULL LOAD CONDITION

LBP = 769'
B = 90'
D = 53'
T = 33'
Δ = 42163 tons

SECTION PROPERTIES

SECTION	LENGTH (ft)	MOD. OF ELAS. (ton/ft²)	MO. INERTIA (ft⁴)	MASS DENSITY (ton-sec²/ft²)
1	9.890000E+01	1.928600E+06	5.710000E+03	8.939000E-01
2	7.690000E+01	1.928600E+06	8.510000E+03	2.235200E+00
3	7.690000E+01	1.928600E+06	1.309700E+04	4.133700E+00
4	7.690000E+01	1.928600E+06	1.578000E+04	5.352900E+00
5	7.690000E+01	1.928600E+06	1.578000E+04	5.470600E+00
6	7.690000E+01	1.928600E+06	1.578000E+04	5.501200E+00
7	7.690000E+01	1.928600E+06	1.578000E+04	5.425200E+00
8	7.690000E+01	1.928600E+06	2.836500E+04	4.303800E+00
9	7.690000E+01	1.928600E+06	1.994000E+04	2.734500E+00
10	9.890000E+01	1.928600E+06	6.460000E+03	1.554300E+00

SECTION	ELAS. FDN (ton/ft²)	ROT. ELAS. FDN	SHEAR AREA (ft²)	POISSON RATIO
1	4.660000E-01	0.	4.000000E+00	3.000000E-01
2	1.534000E+00	0.	6.800000E+00	3.000000E-01
3	2.296000E+00	0.	8.400000E+00	3.000000E-01
4	2.541000E+00	0.	8.400000E+00	3.000000E-01
5	2.571000E+00	0.	8.400000E+00	3.000000E-01
6	2.571000E+00	0.	8.400000E+00	3.000000E-01
7	2.571000E+00	0.	8.400000E+00	3.000000E-01
8	2.532000E+00	0.	7.270000E+00	3.000000E-01
9	2.133000E+00	0.	4.330000E+00	3.000000E-01
10	1.032000E+00	0.	3.000000E+00	3.000000E-01

TABLE 23 - VARIATION OF PROPORTIONS AND RESPONSE FOR CONTAINERSHIPS - FULL LOAD CONDITION

α (degrees)	L (ft)	B/T	Δ (tons)	I (ft^4)	$BM_V \cdot 10^{-3}$ (ton-ft)	$BM_L \cdot 10^{-3}$ (ton-ft)	$BM_T \cdot 10^{-3}$ (ton-ft)	Defltn (ft)	L/B	$\frac{L^2}{BI}$	ω_1 (radian/sec)	V (ft/sec)	σ_b (psi)
180	625	2.754	29,878	13,750	778	209	37.6	0.9872	6.944	400.8	5.617	38.846	24,465
180					221			0.2902					6,943
180	769	3.212	49,650	18,585	710	360	27.0	1.0000	7.254	477.8	3.711		18,940
180					129			0.3022					6,577
180	769	2.727	42,163	15,780	850	470	44.0	1.4099	7.847	586.2	3.792		23,270
180					395			0.6620					10,814
180	875	2.727	49,800	15,780	1071	742.3	45.8	0.8197	9.722	759.0	3.276	38.001	29,320
180					148.4			0.2747					4,062

6.0 COMPUTATION RESULTS

The seaway responses of the four parent ships were computed for 12 headings, for 0° to 180° in 15° increments, and for three significant wave heights. Results are included in Figures 38 to 45. For the variations of the parent ship, computations were made only for those headings where high values of responses were anticipated. For example, if the maximum wave moment occurred at 45° heading for the parent ship, then computations were made for 30°, 45°, 60° and 180° for the variations. Head seas cases were calculated for all parents and variations to assess the possibility of springing.

Calculated results are summarized in the following tables and figures.

Parent Ship	Tables	Figures
STEWART J. CORT (Great Lakes Ore Carrier)	8, 9	5 - 12
"T10" Tanker	13, 14	13 - 20
"C4" General Cargo	18, 19	21 - 39
"C6/C8" Containerships	23	30 - 37

It should be noted that ship displacements in Tables 5, 10, 15 and 20 may differ from values given in Tables 8, 9, 13, 14, 18, 19, 20 and 23. Values in the latter tables reflect actual loading conditions studied and include inaccuracies inherent in the iterative procedure for balancing the ship on a wave.

7.0 DISCUSSION OF METHODOLOGY AND RESULTS

7.1 Methodology

The responses of the four types of ships in a seaway have been calculated using the best methods currently available. The effects of ship proportions have been obtained and plotted in Figures 1, 5 through 37 and 46, and summarized in Tables 17 through 24.

Theoretically, both the deflection and bending moment are affected by the following four factors:

(1) Hull proportions,
(2) Ratio between ship encounter frequency and the two-node frequency,
(3) Ratio between wave length and ship length,
(4) Heading angle and wave height.

Unless the other factors are constrained to small variations, the effects of variation of proportions alone cannot be shown explicitly. In such cases, however, the responses may be beyond the range of interest. In any case, the effects of the variation of ship proportions on the maximum responses is of greatest interest. To obtain this information all the above four factors must be varied in a multiple-dimension space.

Because of the limited scope of this study, this approach is not feasible. As a compromise, the problem has been separated into two phases.

First, the conditions associated with the maximum bending moment of the rigid ship were searched by using the seakeeping program. Then the sea loads at these conditions were used as input to the vibration analysis as described in Section 4.5. As noted earlier, this approach is only approximate because the maximum seaway loads from the seakeeping program are not entirely valid because of the flexibility of the ship's hull.

7.2 Results

The results obtained from the above methods are in generally good agreement with the requirements of ABS. The relations obtained in the study, as shown in Figures 1 and 46, may be useful for design purposes.

For presentation of the results, various relations between the ship proportions and the responses were tried. Only three

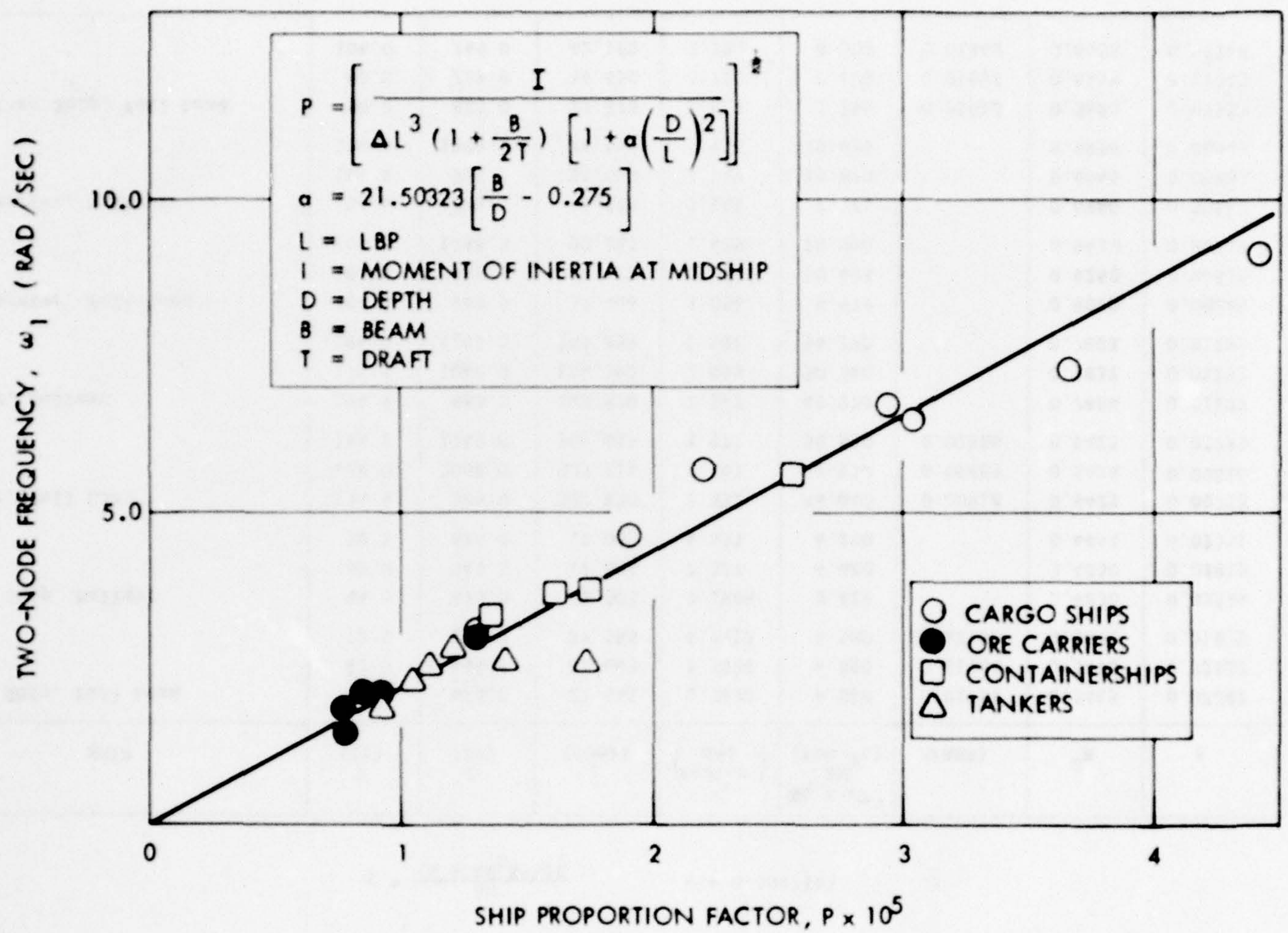

FIGURE 46 - EFFECT OF SHIP PROPORTIONS ON THE HULL FLEXIBILITY (REPRESENTED BY THE TWO-NODE FREQUENCY)

TABLE 24 - RELATION BETWEEN SHIP PROPORTIONS AND HULL FLEXIBILITY FOR THE TWO-NODE FREQUENCY

$$\beta = \frac{BM}{(\Delta + kC_b B^2 L)L} \qquad k = 0.0097143$$

SHIP	B (ft)	L (ft)	Δ (tons)	ω_1 (radian/sec)	$BM_v \times 10^{-5}$ BM_v (ton-ft)	β(ABS)	C_B	β
Cargo Ship, Full Load	94.0	475.0	22,641	6.7670	4.880	0.01751	0.5465	0.02287
	82.0	544.5	22,643	5.7900	4.890	0.01964	0.5470	0.02133
	72.5	614.0	22,585	4.7212	4.580	0.02182	0.5482	0.01875
Cargo Ship, Ballast	94.0	475.0	12,003	9.1986	5.419		0.4830	0.03599
	82.0	544.5	12,004	7.327	4.480		0.4830	0.02819
	72.5	614.0	12,003	6.535	4.260		0.4844	0.02551
Tanker, Full Load	194.9	968.0	302,850	2.537	44.040	0.00818	0.8425	0.00753
	178.0	1060.0	303,114	2.347	49.930	0.00889	0.8424	0.00815
	164.5	1147.0	304,017	1.977	50.530	0.00926	0.8425	0.00789
Tanker, Ballast	194.9	968.0	143,870	2.537	48.270		0.7804	0.01179
	178.0	1060.0	144,392	2.895	50.540		0.7832	0.01192
	164.5	1147.0	143,859	2.582	54.770		0.7803	0.01259
Ore Carrier, Full Load	104.6	788.5	57,486	3.058	8.939		0.9070	0.00849
	104.6	988.5	74,472	2.107	10.493		0.9260	0.00618
	104.6	1288.5	90,267	1.425	20.990		0.9430	0.00742
Ore Carrier, Ballast	104.6	788.5	45,645	3.141	7.074		0.6960	0.00862
	104.6	988.5	54,639	2.149	10.870		0.6646	0.00883
	104.6	1288.5	74,549	1.926	20.620		0.6956	0.00942
Container Ship, Full Load	90.0	625.0	29,878	5.617	7.786	0.01920	0.5690	0.02153
	90.0	769.0	49,650	3.710	7,100	0.01697	0.6459	0.01040
	106.0	769.0	42,163	3.790	8,500	0.01840	0.6458	0.01146

non-dimensional parameters can produce meaningful relations, namely L/B, $L^2/BI^{\frac{1}{4}}$, $BM/(\Delta + \Delta')L$. The effects of these parameters are discussed in the following paragraphs.

Based in the calculated results, the following observations can be made regarding the effects of variation of ship proportions:

(1) No obvious effects of B/T on any response can be found for constant displacement of a given ship,

(2) All responses are affected by the L/B ratio and by $L^2/BI^{\frac{1}{4}}$. However, increase of these proportions does not necessarily increase the response, especially when L/B is small.

(3) The effects of all ship proportions on the maximum vertical vibration bending moment can be obtained from Figure 46 and the following simple relations:

$$BM_V = 0.004\ (\Delta + \Delta')\ L\ \omega_1$$
$$\Delta' = 0.0097143\ C_B B^2 L \qquad (17)$$

For any ship with given proportions the two-node frequency can be obtained from Figure 46. With ω_1, the maximum vertical vibration bending moment can be calculated from the above equation.

The maximum vertical bending moment was calculated for a significant ocean wave height of 25 feet and 20.5 feet for the Great Lakes.

(4) The above equations indicate that the bending moment is proportional to displacement, added displacement due to the water, block coefficient, square of the beam, square of the length, and the two-node frequency.

(5) Since the hull flexibility is inversely proportional to the frequency, the bending moment for a given ship and loading condition decreases with increase in flexibility.

(6) Of particular interest are the results obtained for the Great Lakes ore carrier STEWART J. CORT given in Figures 5 through 7. Results show that responses for the springing condition for a 5 foot-high significant wave can be higher than the responses for the ship on a 20 foot-high significant wave.

8.0 CONCLUSIONS, APPLICATIONS, AND RECOMMENDATIONS

8.1 Conclusions

From the previous discussions and the results of the vibration calculations for the four ship types, the following conclusions can be made:

8.1.1 Hull Flexibility - In the past, many shipboard vibration problems have been attributed to the trend toward increasing hull flexibility. For this reason, it followed that a criterion for a limit to hull flexibility was needed. However, from the results shown in Figure 1 and 46, it can be concluded that a specific limit to hull flexibility, with particular respect to total bending moment, may not be necessary for the following reasons:

(1) Even though the calculations and results reported herein are subject to the limitations discussed earlier, the relation showing the bending moment for a given ship and loading condition decreasing with increase in hull flexibility is valid (Figure 1). Obviously, a completely flexible ship in waves cannot be subjected to any bending moment. For example, naval architects have seriously proposed hinged ships, to reduce bending moment by increasing hull flexibility.

(2) Almost all shipboard vibration problems are local problems. The following are often mentioned examples:

Propulsion system problems occur because of hull flexibility and corresponding hull-shafting-bearing system interactions. The ship hull must provide a foundation stiff enough for the shafting system and machinery. This problem can be solved by reinforcing the portion of hull involved. Machinery compartments of large vessels are generally located well aft. In these cases, the after one-fourth or one-fifth of the hull length can be reinforced to the desirable degree. This only affects the hull flexibility slightly. In this case, a very flexible hull, with proper support of the machinery and shafting system, can still be acceptable.

For special ships such as LNG carriers, hull deformations can cause problems in way of LNG containment. The degree of flexibility that can be tolerated generally, or locally, depends upon the nature of the containment system, including the choice of independent tank versus integrated containment systems.

- Hull Flexibility can be a cause of hull vibrations which cause habitability problems in personnel spaces. Again this is a local problem which can be solved locally and is not necessarily related to hull flexibility.

(3) Within the accuracy of the existing ship vibration and seakeeping theories, the vertical bending moment seems to be decreasing with decrease in the ship hull two-node frequency. This implies that bending moment decreases with increase in hull flexibility when springing is not a factor.

8.1.2 Methodology - The ship motion and ship vibration problem is essentially a hydroelastic problem. The existing methods based on combining rigid-ship seakeeping theories and flexible-ship vibration theories may lead to unacceptable errors for flexible ships, which is the general case. With rigid hull girders, the vibration problem reduces to allow use of the existing seakeeping theories.

Because of the uncertainties in the damping and the forward-speed effects, and the hydroelastic effects on the ship response, the existing methods of analysis, including the one used in this study, are not adequate for springing calculations.

8.1.3 Seakeeping Theories

(1) The assumption of rigid-ship hulls inherent in all existing seakeeping theories is not valid for large ships because of the effects of the hull flexibility.

(2) Even for relatively small, stiff ships the existing seakeeping theories do not properly account for the forward-speed effects and the hydrodynamic coefficients toward the ends of the ship.

(3) In high-energy waves, where the two-node frequency of the ship is much higher than the encounter frequency, the seakeeping theories tend to over estimate the seaway loads.

(4) In low-energy waves, where the two-node frequency of the ship is close to, or coincides with, the encounter frequency, the seakeeping theories tend to over estimate or under estimate the sea loads.

8.1.4 Vibration Theories

(1) Since a hydroelastic formulation of the vibration theory is beyond the scope of this project, all equations of motions in this report are tentative and should not be used directly. The complete set of equations of motion

has not been formulated. For this reason, the exact expressions for the terms associated with the forward speed have not been formulated. At this time, it is only certain that some important terms for forward-speed effects have been ignored in the existing vibration theories and the exact expressions of these terms have not been established.

(2) The forward-speed effects on the vibration response are important. However, different investigators still use different terms for the forward-speed effects. It is evident that many significant forward-speed effects have been ignored in the existing ship vibration theories.

(3) Forward speed has the following important effects on the vibration characteristics of ships:

- Natural frequencies of the ship hull.
- Linear damping of the vertical motion of the ship section.
- Rotary damping of the ship section.
- Hydrodynamic excitation force upon the ship's hull.

Note that in many existing ship-vibration methods only the effects on linear damping have been considered.

8.1.5 Damping - The damping coefficients and added mass from the seakeeping programs are, in general, quite accurate for most of the ship's hull. However, this accuracy decreases toward the ends.

Structural damping of ships is still an unsettled subject because of the lack of reliable data. Methods used for full-scale damping tests are generally inadequate.

8.2 Applications

The major study results that may have future application are presented in Figures 1 and 46. An analytical expression of the line shown in Figure 1 is given in Equation (17). The line given in Figure 46 can be represented by the expression:

$$\omega_1 = 218{,}000\ I^{\frac{1}{2}} \left\{ \Delta L^3 \left(1 + \frac{B}{2T}\right) \left[1 + 21.5032\left(\frac{B}{D} - 0.275\right)\left(\frac{D}{L}\right)^2\right] \right\}^{-\frac{1}{2}} \qquad (18)$$

where

- I = moment of inertia of the ship in ft^4
- Δ = displacement, long tons
- L = LBP, feet
- B = breadth, feet
- D = depth, feet
- T = draft, feet
- ω_1 = two-node frequency

The effects of varying ship proportions can be evaluated by using Equations (17) and (18), as illustrated in the following example:

The tanker UNIVERSE IRELAND has been selected for this example calculation since some experimental and calculated vibration data has been published in Reference 20.

The UNIVERSE IRELAND is a 326,000 DWT tank vessel with the following principal characteristics:

LBP	1075.94'
B	174.83'
D	104.99'
T	81.417'
C_B	0.86
Δ	375,811 long tons
I	216,483 ft^4

8.2.1 Estimation of Two-Node Frequency - The two-node frequency can be estimated by substituting the ship proportions into Equation (18). From Equation (18) we have:

$$\omega_1 = 2.8724 \text{ rad/sec}$$

The value ω_1 calculated by the American Bureau of Shipping is 3.09 rad/sec for the full-load condition (Reference 20). The difference is about 7%. Since the ABS computer program has been validated by many full-scale measurements, the value of 3.09 rad/sec should be accepted as quite accurate.

8.2.2 Effect of Breadth Variation - To evaluate the effect of varying ship breadth, all parameters with the exception of draft were held constant in this example. The draft was allowed to change to maintain displacement constant. In such a case Equation (18) reduces to:

$$\omega_1 = \frac{4.68815}{\sqrt{(1 + 3.51267B^2 \times 10^{-5})(0.943694 + 0.0019505B)}}$$

If now breadth is increased from 174.83' to 200', the ω_1 value becomes

$$\omega_1 = 2.61754 \text{ rad/sec}$$

This value of the two-node frequency is less than the 2.872 rad/sec calculated for the 174.83'breadth ship. Accordingly, the wider ship is more flexible.

Variation in other ship parameters such as length, depth, draft and displacement can be evaluated in a similar manner using Equation (18).

8.2.3 Effect of Material Changes - If high-strength steel is used, the classification societies usually allow certain reductions in the section modulus. If all other ship parameters are held constant, the effect on hull flexibility can be readily calculated.

It is assumed that the original deck and bottom plating of the tanker was constructed of mild steel. If a high-strength steel with $\sigma_y = 34{,}000$ psi, $\sigma_u = 66{,}000$ psi was substituted for the bottom and deck plating, then the section modulus can be reduced to

$$SM_{hts} = \frac{70900}{(34000 + 44000)} \times SM$$

$$= 0.908974 \text{ SM}$$

From Equation (18) we have

$$\omega_{1,hts} = 0.95340\ \omega_1$$

For other materials or considerations of materials, the method of equivalent moment of inertia is given in Section 4.6. Following the calculation of the equivalent moment of inertia, Equation (18) should be used to obtain the two-node frequency.

8.2.4 Estimation of Total Vertical Bending Moment - The maximum vertical bending moment that a ship may encounter can be calculated by using Equation (17) or Figure 1. For the example tanker

$$\Delta' = 274{,}745$$

$$BM_V = 0.004 \times 650556 \times 2.8724 \times 1075.94$$

$$= 8{,}042{,}251 \text{ ft-ton}$$

The above bending moment appears to be excessive. However, as indicated throughout this report, the bending moment predicted by Equation (17) or Figure 1 may be too conservative because of the many uncertainties involved in the existing theory. The development of a more accurate theory as recommended in Section 8.3, would result in appropriate modification of Equation (17).

8.3 Recommendations

The scope of the study report herein was necessarily limited. Accordingly, the following specific areas of investigation are recommended for future studies:

8.3.1 Development of a Computer Program Based on a Hydroelastic Formulation - All elements for the development of a computer program for study of ship vibrations based on a hydroelastic formulation are available. The potential benefits to be derived by such a program include the following:

(1) The computer program could be used to evaluate the various methods available for investigating the effects of forward speed on natural frequencies, damping, and the excitation force on the ship. The best method could then be selected for future use.

(2) The program could be used to determine the error introduced by rigid-body seakeeping theories.

(3) Damping experiments should be guided by theory, and availability of a more accurate vibration theory will improve results. For example, existence of such a theory would permit the isolation and verification of the components of damping forces.

(4) The computer program could be used to verify the relations given in Figures 1 and 45.

8.3.2 Generalization of the Analytical Approach to Hull Flexibility - For the purpose of this study, four specific ships were selected for analysis. The ship vibration problem is complex and the vibration analysis is costly. In order to properly determine the general relationships between ship proportions and vibration responses, within reasonable limits of time and budget, the following approach is recommended for future studies:

(1) All structural and hydrodynamic coefficients in the equations of motions should be treated as functions of ship proportions in appropriate expressions rather then as simple numerical values.

(2) By defining the hull geometry by simple but realistic mathematical expressions, the solution for vibration response could be expressed in terms of the ship proportions.

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